EVAN

LEARNING OUTCOME 1

Assessment:		Short answer, practical exercise.
Performance:	a.	List the general distribution requirements for ventilation/air conditioning system applications.
	b.	Inspect two different ventilation systems and state how they do or do not meet the above requirement.

REQUIREMENTS

- Basic Human Comfort
- Duct Work Construction
- Site Work
- Necessary Skills
- How to Install
  - new buildings
  - existing buildings
- Spot Cooling/heating
- Return Air
- Evaporative Systems
- Refrigerated Systems
- Basic layout of system and compatibility with other mechanical services.

Suggested teaching time: 4 hours.

VENTILATION NR13

LEARNING OUTCOME 1

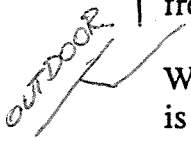
THE PROCESSES OF AIR CONDITIONING

Different levels of comfort for occupants of buildings can be achieved by processes varying from the opening of windows to the full air conditioning by mechanical processes. This chapter deals with the principles of natural and mechanical ventilation, evaporative cooling and full air conditioning, whilst the remainder of the text deals in detail with the mechanical processes most extensively used by Australian engineers.

'Discomfort' results from extremes of temperature (for which the only solution is heating or cooling) and from 'stuffy' conditions which can result from poor air movement, high humidity and concentration of odours or smoke. Ventilation can usually provide the remedy.

NATURAL VENTILATION

The prime purpose of ventilation is to remove objectionable air and to replace it with fresh air.



When outside air is more acceptable than room air, the simplest way to improve comfort is to open doors or windows, but the degree of relief will depend upon a number of factors:

a) The size and type of windows.
b) The location of the openings.
c) The velocity and direction of the prevailing wind.
d) Window treatments, such as fly wire, curtains and blinds.

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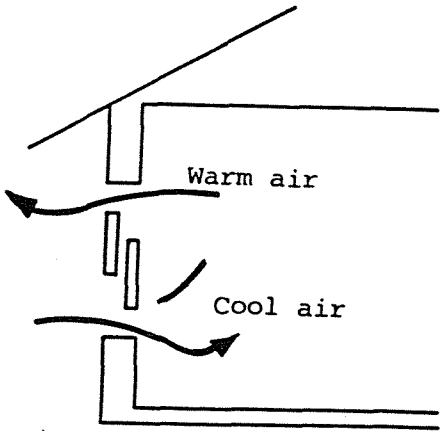
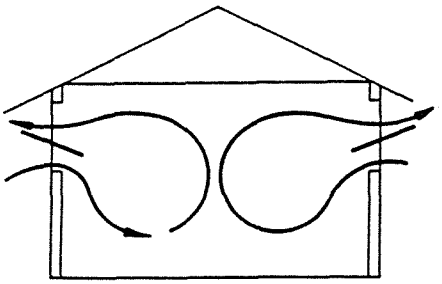


Fig. 19.1

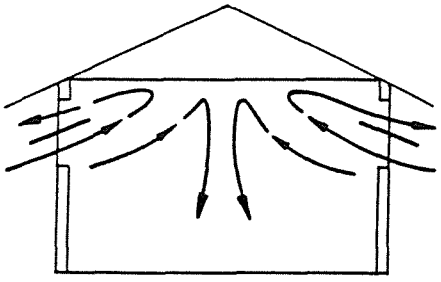
Natural air flow induced by double-hung sash windows.

Other windows are more effective when they can be turned into the direction of the breeze, such as side-hinged casements.

Side-pivoting windows and louvres as shown in Figure 19.2 are most effective when used as shown for summer and winter use.



General air flow pattern for pivoting windows - summer ventilation.



General air flow pattern for pivoting windows - winter ventilation.

Fig. 19.2

LOCATION OF THE OPENING

Efficient ventilation requires openings on opposite sides of a building where a maximum use can be made of the prevailing breezes, and the high and low pressures which result on the windward and lee sides of the building because of the wind forces. Little effective movement is possible when the openings are on one side, with openings on adjacent walls only marginally better. Figure 19.3 shows the direction of air movement due to wind.

VELOCITY AND DIRECTION OF THE PREVAILING WIND

The value of openings in opposite walls has been stated, but the effect of increased velocity of the wind is such that constant regulation of the size of the opening is necessary to prevent excessive draughts and even damage inside the room. Changes to wind direction can result in an acceptable level of ventilation becoming unacceptable or draughty. Figure 19.3 shows how the wind force causes good ventilation with windows on windward and lee sides.

WINDOW TREATMENTS

Because the changes in air pressure are very small, any resistance to free air movement greatly reduces the air flow. Thus the effect of flywire and heavy drapes will be to greatly reduce movement unless the wind pressure can overcome this resistance.

The major disadvantage of natural ventilation then, is that it is unreliable, being subject to weather, wind, and the normal requirements of building design.

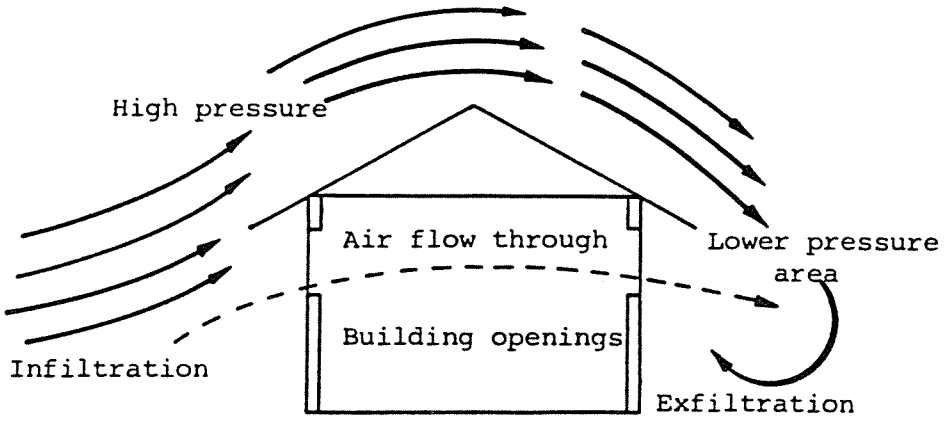


Fig. 19.3
Effect of wind forces on a low building.

MECHANICAL VENTILATION

Early methods to provide ventilation in an attempt to make life more bearable during oppressive weather conditions, may seem humorous today; however the 'punka', as used in India, moved air by displacement, as shown in Figure 19.4. It consisted of a large, manually-operated fan, generally hung from the ceiling, with movement was achieved by pulling a rope.

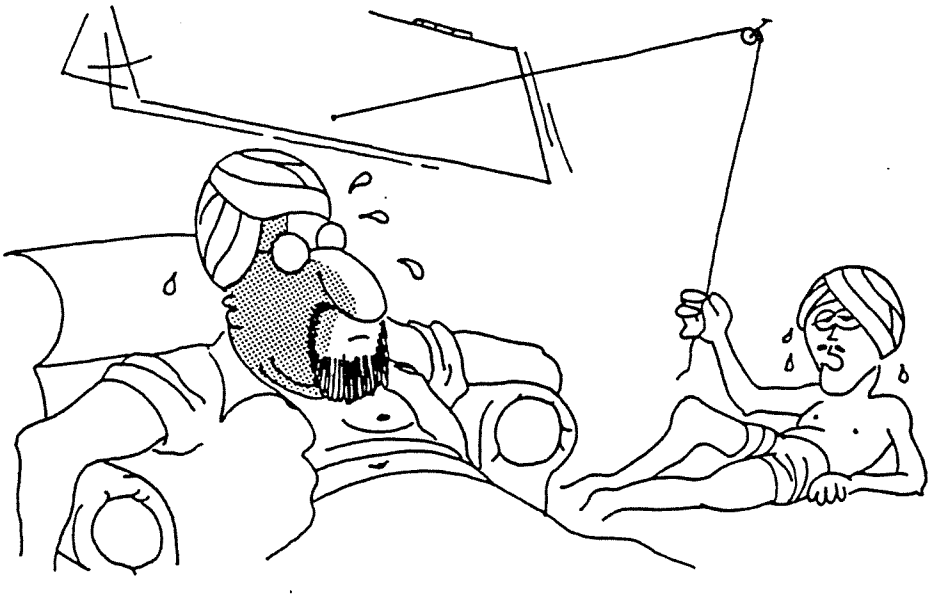
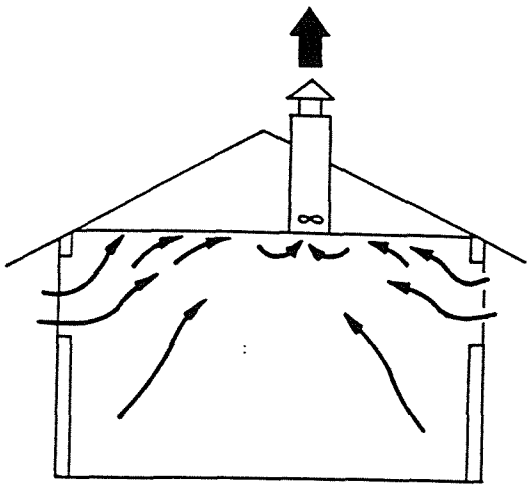


Fig. 19.4

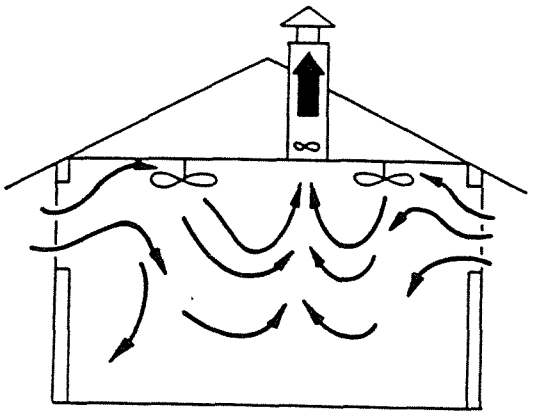
Various other mechanical methods were quickly followed by more sophisticated and automated equipment. Fans of varying shape and size were driven via a shaft by such methods as water-flow over paddle wheels, by large clock mechanisms or by pre-loaded weights which when permitted to unwind, which would drive a gear and eccentric which was attached to a moveable blade. Unfortunately all these alternatives where rather cumbersome and required various alteration to the building or large amounts of time and energy to operate.

However, these early attempts did fulfil their purpose and today we use the same principle of moving air to effect personal comfort, convenience or safety. Now air is moved mainly by means of electric motors which drive propeller or centrifugal fans.



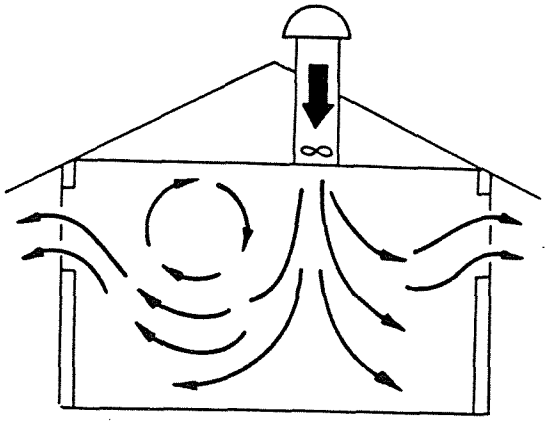
(a) Induced ventilation.

Fresh air entry to room is at high level, above occupant height mainly.



Fresh air entry is forced down and mixed with room air at occupant level.

(b) Induced ventilation with circulation fans.



Air release through door grilles, windows, etc.

(c) Forced ventilation system pressurised air supply to all points.

Fig. 19.5

Examples of mechanical ventilation are found in buildings used for every day activities, where toxic fumes are not a problem.

Mechanical ventilation may take several forms today, depending upon the application. Generally, these systems may be classified as:

(a) *Forced ventilation,* where air is forced into the room by a fan, and allowed to leak out through doors and windows.

(b) *Induced ventilation,* where a fan, mounted either in the ceiling or high on a wall, exhausts air outside allowing fresh air to enter via doors and windows.

(c) *Combined forced-induced ventilation,* with inlet and outlet fans.

Applications usually requiring ventilation are:

(a) Buildings and rooms occupied by people at work.

(b) Machine and plant rooms where heat is generated.

(c) Process plants requiring quick cooling of foods, confectionery, print, etc.

(d) Areas where toxic or unpleasant fumes can accumulate.

Figure 19.5 shows an application of ventilation to large public buildings such as halls, schools, workshops and offices. Similar ventilation may be applied to any room where heat is generated, but *induced ventilation* is generally preferred to forced ventilation where heat and fumes could be blown back over occupants.

Laboratories handling particularly toxic or radio-active materials often required specially-designed fume cupboards made of non-corrodible plastic and special glass covers on all sides.

Comfort of the workers is usually a secondary consideration in industrial processes, with safety the prime objective. However, when people are involved, consideration must be given to:

- High air noise levels,
- Draughts,
- Air turbulence,
- Temperature,
- Moisture content,
- The purity of the circulated air.

DO'S AND DON'TS WITH VENTILATION

- Where only windows are provided, make sure both top and bottom openings are used.

- Where possible incline windows for downward deflection of air in summer and upward deflection in winter.
- Do not use screens or heavy curtains.
- When using ventilation fans open the bottom windows or door to admit air at low level.
- Regularly clean any air supply and exhaust air grilles.
- Do not run ventilating fans with doors and windows shut. Circulating fans may be used, but they will not provide ventilating air.
- Leave ventilating fans on overnight in hot weather to cool the building ready for the next day.

AIR CONDITIONING DEFINED

Air conditioning is often referred to as a science.

"The Science of Supplying and Maintaining a Desirable Internal Condition Regardless of External Conditions."

This may be incorporated in a machine which heats, cools, cleans and circulates air and controls the moisture content in simultaneous processes, all year round. Complete air conditioning is sometimes referred to as **Climate-controlled Conditions,** and is generally applied to large offices and public buildings. This type of conditioning provides a high level of control of all the requirements for human comfort.

Very few of the systems used to provide the fully conditioned air are identical because of the particular design of each building. While the refrigerating and heating plants may be the same, the methods used to distribute cool or heated air differ because of the requirements of the buildings.

The main types of buildings requiring complete air conditioning are:

- High-rise apartment buildings,
- High-rise office blocks,
- Multi-storey departmental shops,
- Theatres and entertainment centres,
- Large shopping complexes,
- Hospitals.

The applications of the air conditioning process are more wide ranging than most people realise. Two extreme applications for comparison could be; air conditioning of astronauts' space suits for life preservation, and the climate controlled conditions for mushroom cultivation. Other processes will become apparent as you learn more about this industry. We will now investigate the factors to be controlled in an *air conditioning process*. These are:

- Control of Air Temperature.
- Control of Air Humidity.
- Control of Air Cleanliness.
- Control of Air Purity.
- Control of Air Distribution

To achieve complete air conditioning, equipment must be provided, together with the necessary controllers, to achieve the desired internal conditions, under all prevailing external internal loads.

Equipment to achieve full air conditioning must therefore provide for the:

- Heating,
- Cooling,
- Humidifying,
- Dehumidifying,
- Filtering,
- Purifying, and
- Distributing

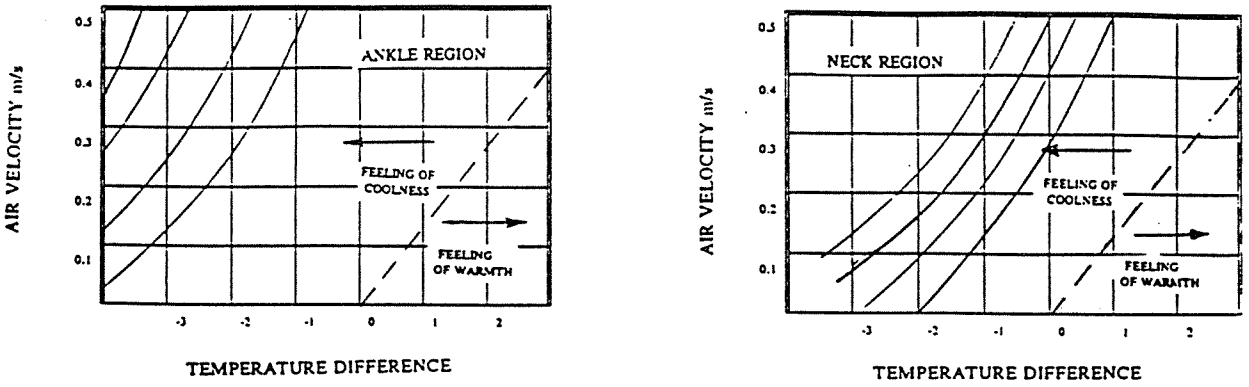
of air supplied to the occupied areas of the building.

We will now look at the methods employed to produce these effects.

AIR HANDLING AND DISTRIBUTION SYSTEM

In any air conditioning system where duct work and air distribution components are used the correct zone velocities and air movement are as important as the correct cooling and heating system.

The activity of the people in the controlled zone should be taken into account when deciding upon acceptable velocities. The relationship of air velocity and air temperature difference between supply air and zone air is shown, as a percentage of zone occupants who will complain, in the graph on the following page.



Percentage of Room Occupants Objecting to draughts.

EXAMPLE: For the neck region, a velocity of 0.3m/s at a temperature of 0.5 K below zone temperature will be acceptable to 80% of the occupants.

The accepted velocities measured at head height for both heating and cooling is shown in the following table.

Activity	Application	Maximum velocity M/S	
		Heating	Cooling
Long sitting	Office work	0.2	0.1
Short sitting	Restaurants	0.3	0.15
Light work	Shops and light manufacturing	0.35	0.2
Heavier work in war room	Dancing, cooking, heavier manufacturing	0.45	0.3

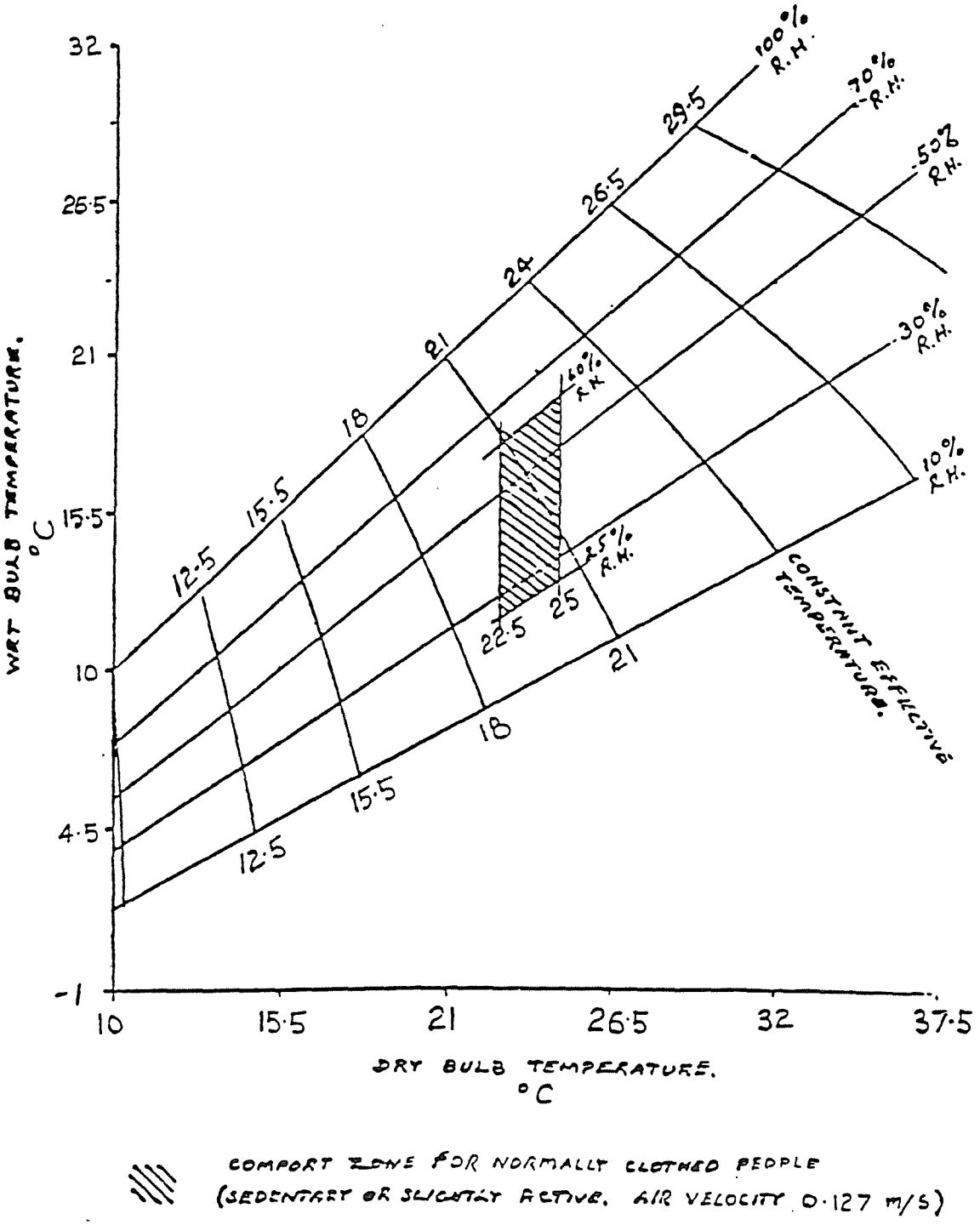
Air velocities below 0.075m/s give a feeling of *stagnation.*

The rate at which a body will dissipate heat by convection and evaporation depends upon the air velocity. Air movement is required to ensure uniform temperature but velocities must be kept within limits or occupants will feel draughts. Maximum air velocity within a zone is usually assumed as 0.21m/s while a minimum velocity of 0.127m/s is necessary to ensure distribution of temperature through the zone.

The degree of comfort experienced by the individual will depend upon:

a) The dry bulb temperature (°C)
b) The wet bulb temperature (°C)
c) The relative humidity
d) The velocity (m/s)

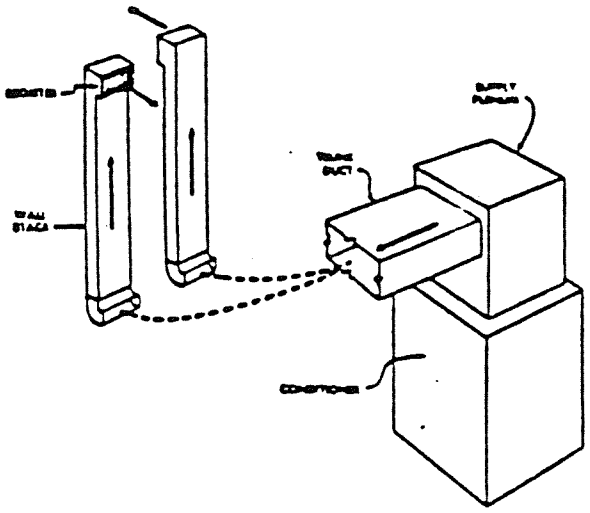
The measure of comfort taking these factors into account is known as the *Effective Temperature.* This can be defined as the temperature of saturated still air which gives the same feeling of warmth or coolness as the condition under consideration.



The effective temperature graph shows what has been determined as the *comfort zone.* For office workers, if all occupants are to feel comfortable, the dry bulb temperature must be between 22.5°C and 25°C. What these lines also tell us is that at 23°C and 55% R.H. and 24°C and 40% R.H. we have the same effective temperature of 21°C. That is, the feeling of comfort or discomfort would be the same at these two conditions.

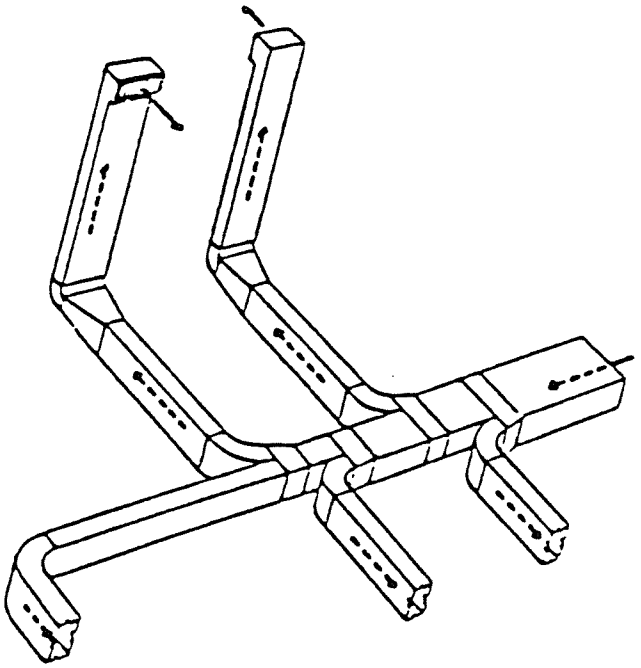
TYPICAL AIR CONDITIONING SYSTEM

The ducted air conditioning system incorporating a spray humidifier is shown in the following diagrammatic sketch.



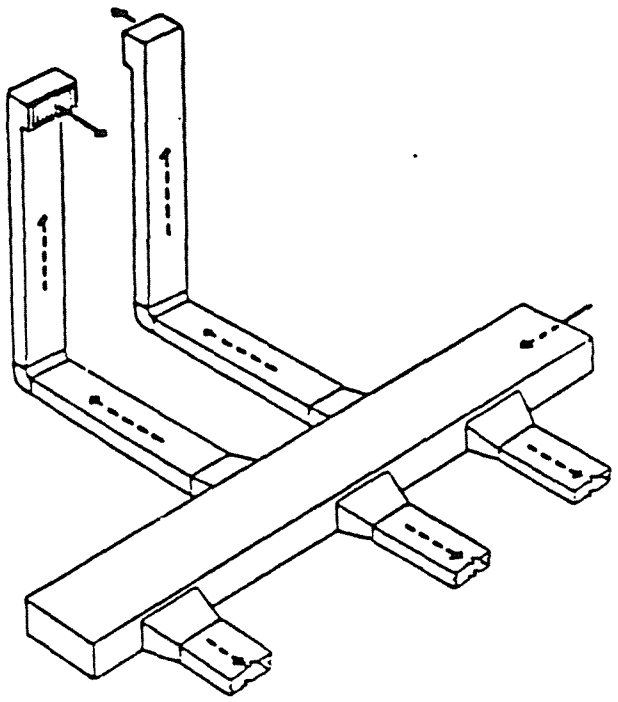
Dotted Lines Represent Duct System
To Be Installed

a) One system of duct layout is the *Trunk* duct system. This system is quite complicated but when properly designed ad installed, provides good control of separate air streams and uses a minimum of sheet metal.



Trunk Duct System

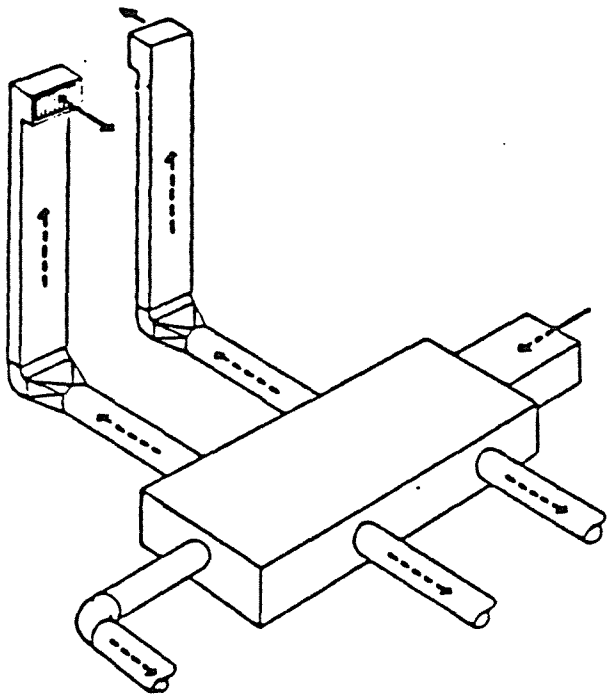
b) Another solution to the same system is the *Extended Plenum* duct system. The main feature being that the trunk duct does not vary in size even after branch ducts have been taken off. Careful measurements of varied shapes of trunk ducts are eliminated. The location of branch ducts can be installed after the extended plenum has been installed. A greater amount of sheet metal will be used in the extended plenum than in the trunk duct with its reducing sections. Often the labour savings more than compensates for the material costs.



Extended-Plenum Duct System

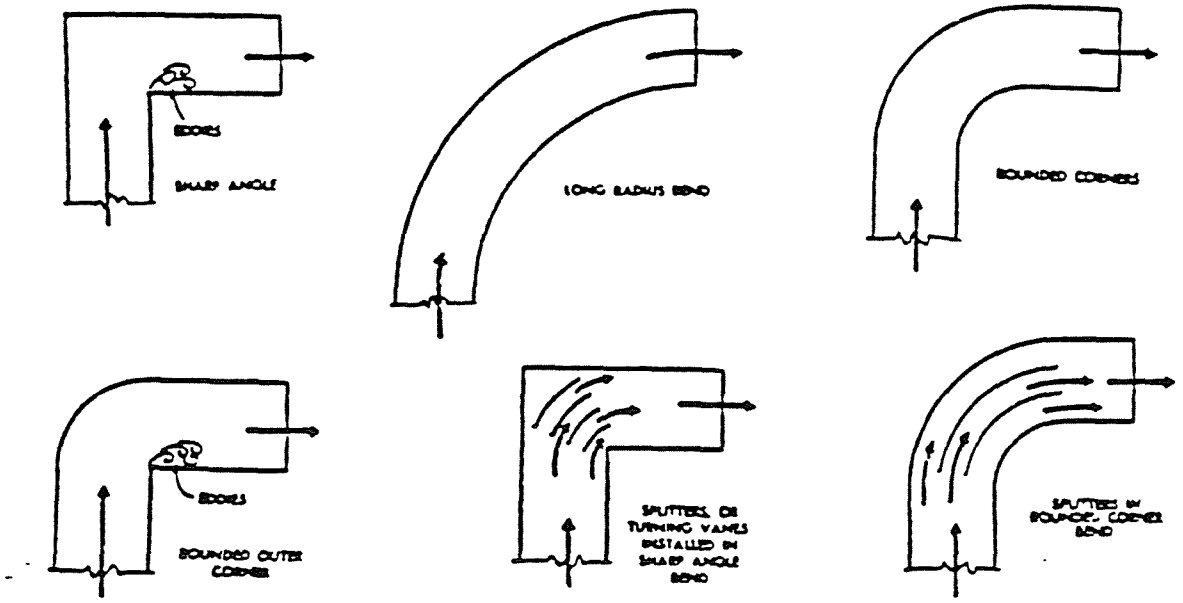
c) A third common duct system which could be used is the *Box Plenum*. The box plenum differs from the extended plenum in the size of the plenum. The plenum box is quite large and air from the conditioner is delivered to this large box, where the initial air velocity is considerably reduced. From this box the air is distributed to the various branch ducts.

The advantage of the box plenum system over the trunk duct system is simplicity in construction and reduced manufacturing labour costs.

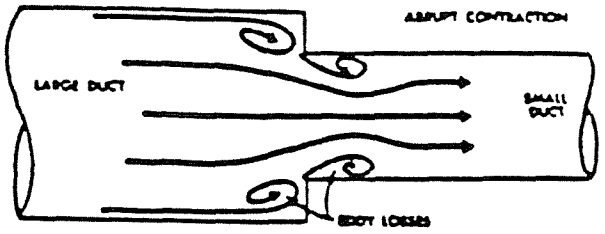


Box-Plenum Duct System

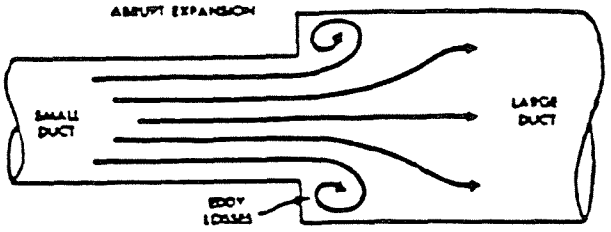
Any duct system is a compromise between the resistance created by air flow and the cross sectional area of the duct in which the air is flowing. To keep the resistance as low as practical, for any given installation some thought must be given to the resistance created by bends, contractions, expansions, etc., in the duct system. Typical bends, contractions and enlargements are shown below and on the following page.



Different ways of making a 90 degree bend.
Some involve greater pressure losses than others.

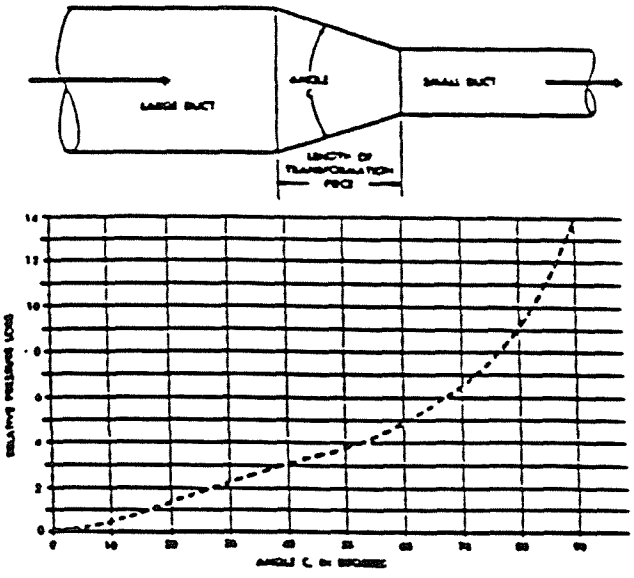


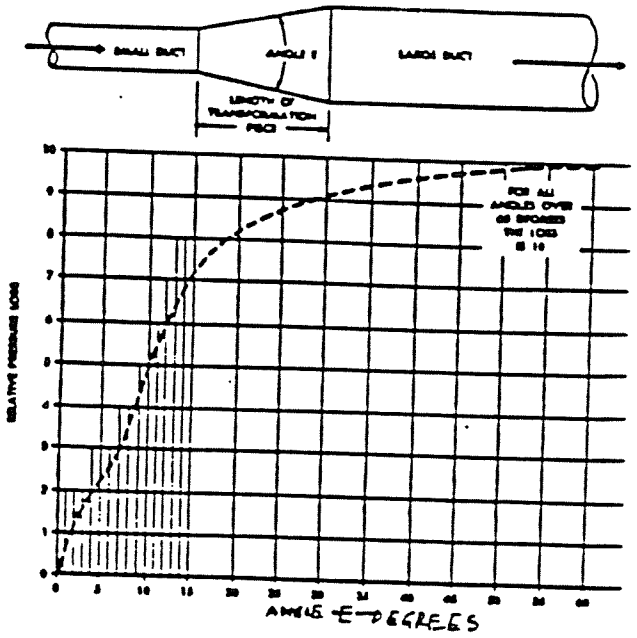
Pressure losses occur with
abrupt reducing fittings



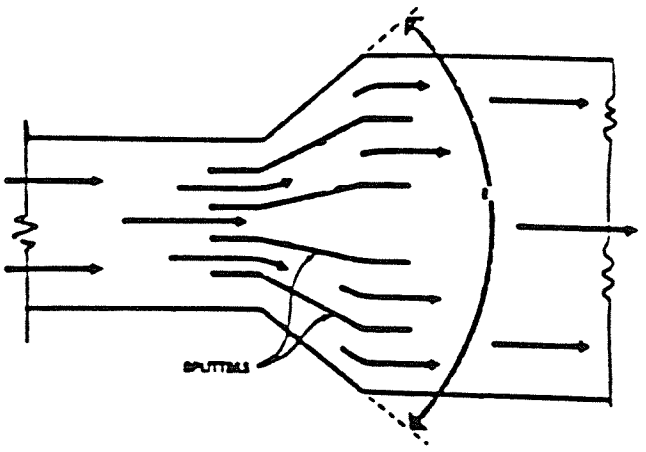
Abrupt expansion results in
excessive pressure losses.

To overcome some of the resistance created by abrupt expansions on contractions these fittings are usually made angular as shown in the following figures.





For sudden contraction in transformation piece with an included angle of approximately 60° is a reasonable compromise and for sudden enlargements an included angle of between 10° and 20° is frequently used. Where space limitation requires greater included angles then splitters similar to those used in bend may be installed.



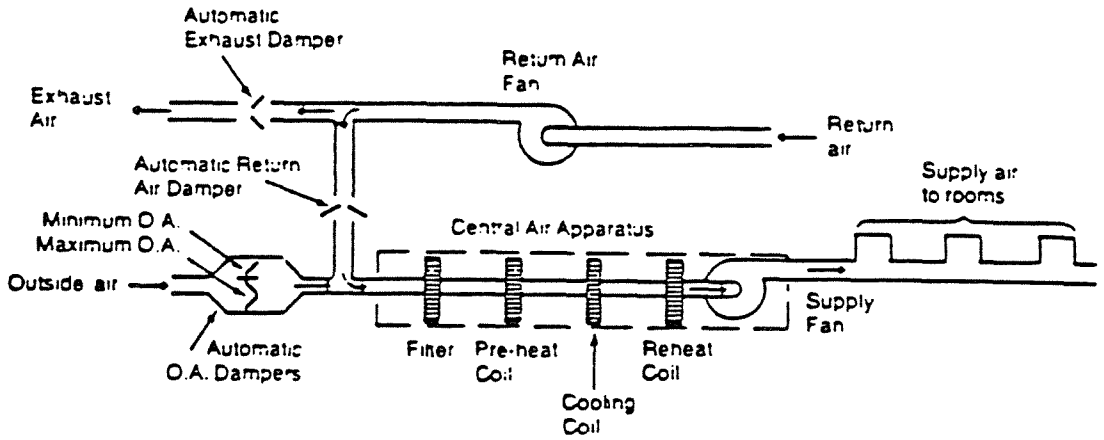
Using Splitters To Reduce
Expansion Angle

To summarise, therefore, the main items contributing towards greater resistance in a duct system are:

a) High air velocities
b) Small cross sectional duct areas
c) Large air volumes
d) Long duct lengths
e) Changes in direction of air flow
f) Contractions in the air stream
g) Expansions in the air stream

TYPICAL AIR CONDITIONING SYSTEMS

SINGLE UNIT (LOW VELOCITY SYSTEM)



Air handling system—single-zone with return air fan and economizer control

This is a simple system with the addition of *economiser control* which allows use of outside air for cooling. Note the two outside air dampers. The two dampers will supply 100% outside air to the system. The minimum O.A. damper will supply required ventilation (normally 10 to 20% of total supply air). The maximum O.A. damper is usually controlled by thermostat and opens gradually to admit outside air when it is cool enough for use in the system.

Provision must be made to exhaust used O.A. and this exhaust damper must be automatically co-ordinate with the return air and maximum outside air dampers.

The economiser control can be used with other types of systems. This is a constant volume system with no individual room control.

The activity of the occupant also contributes to the particular method used to produce the desired result. For example, the proportional mixing of fresh and recirculated air is quite acceptable in an office building but most undesirable in a hospital where germs and bacteria may be transmitted from patient to patient. Similarly, in a laboratory where air purity is critical during drug manufacture or testing of materials, a specific air condition maybe required - to overcome possible variations in measuring devices. On the other hand, largely recirculated air with some fresh air, conditioned to an 'average' comfort level would be acceptable throughout a department store or dry goods shop.

Total Climate Control by air conditioning is a very costly exercise and may be neither feasible nor necessary. Imagine a situation where the worker in a very large factory requires improved environment for health and production efficiency. Personal relief in this case may be achieved by "Spot Cooling".

Unit supplying fresh air, or air conditioned by mechanical or evaporative systems.

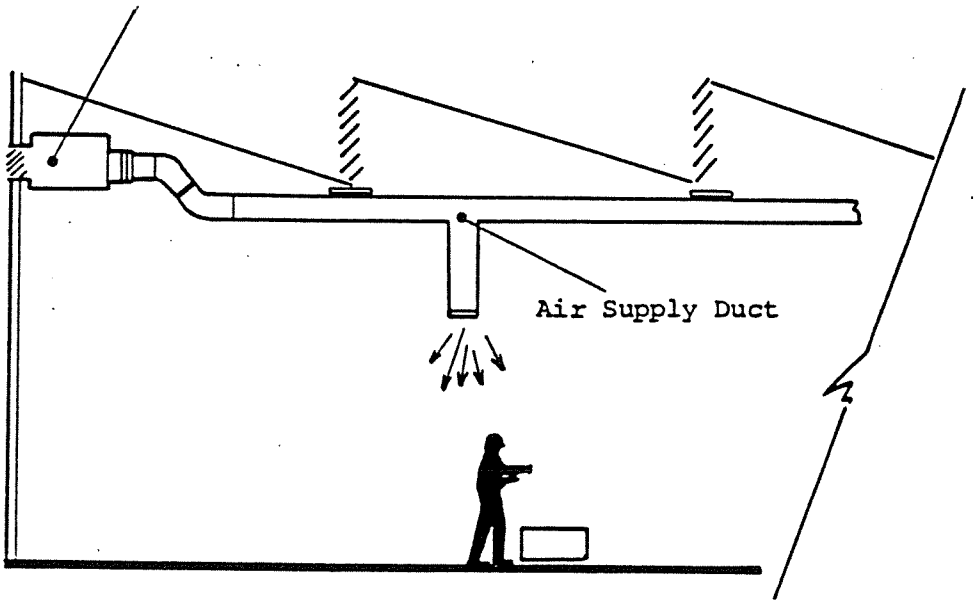


Fig. 19.6

Spot cooling.

Figure 19.6 shows a factory worker exposed to an air supply which is cool, fresh or conditioned by either of the alternatives suggested in the sketch. This air is not returned to the plant, as it is diluted and expelled with the existing factory air. However it does offer a considerable amount of relief to those for whom it is designed. Spot cooling is used extensively in environmental hostile situations, such as smelters and welding booths.

EVAPORATIVE COOLING

This system uses the effect of "Latent Heat" to cool the air as it passes over a water-soaked porous material. The materials vary from plastic compositions to pads of cotton covered straw, and heat is removed (absorbed) from the air in changing some of the water to a vapour.

The amount of cooling of the air depends entirely upon the amount of moisture already in the air in changing some of the water to a vapour.

The amount of cooling of the air depends entirely upon the amount of moisture already in the air. Thus, with dry air in inland areas, the temperature can be reduced by up to 15°C. The system is less effective near the coast, because during this heat transfer process the incoming air will absorb even more moisture, depending upon the original condition of air entering the machine. Therefore, if the entering air is close to saturated, very little cooling by evaporation will occur, and temperature reductions may be offset by the increased humidity within the space.

Because of the increase in relative humidity as air is cooled and moisture added, air should *never* be recirculated through an evaporative cooler. To be effective, only fresh outside air should be brought through to cooler, and be exhausted out through the other sides of the conditioned space. The systems described earlier, and a great many are sold for domestic and small commercial applications.

Figure 19.20 shows an 'exploded' diagram of a typical unit designed for installation in windows. Larger, but similar units, may be installed on roofs, with air ducted to required rooms.

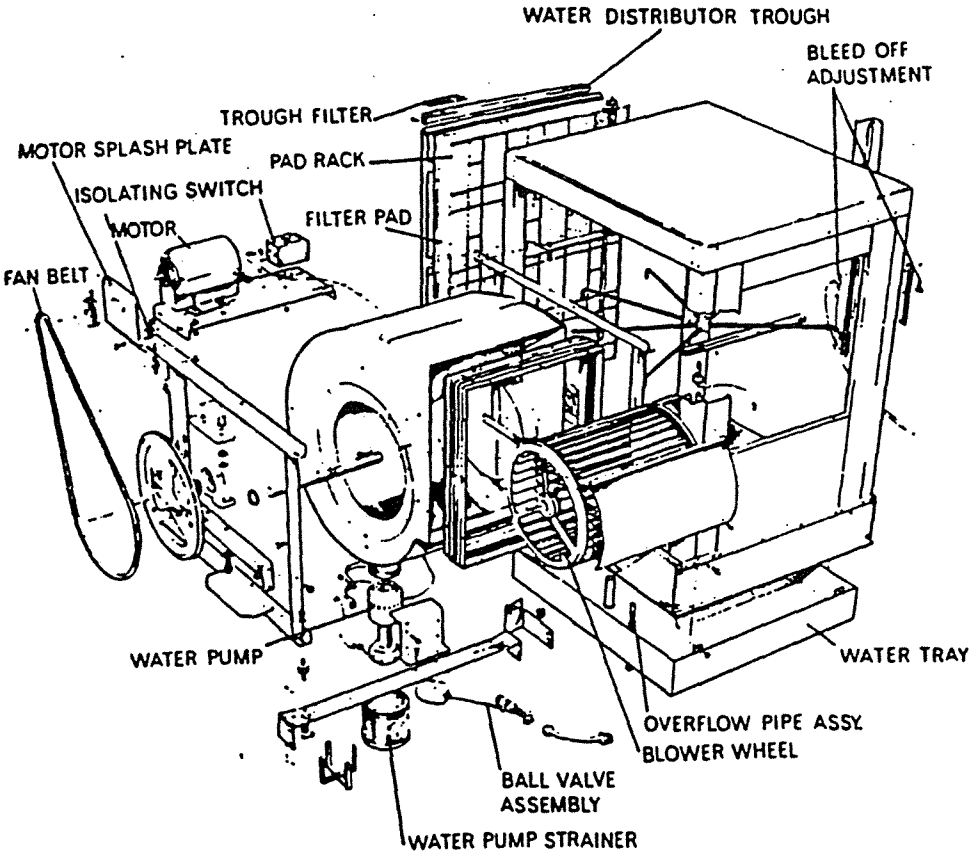


Fig. 19.20

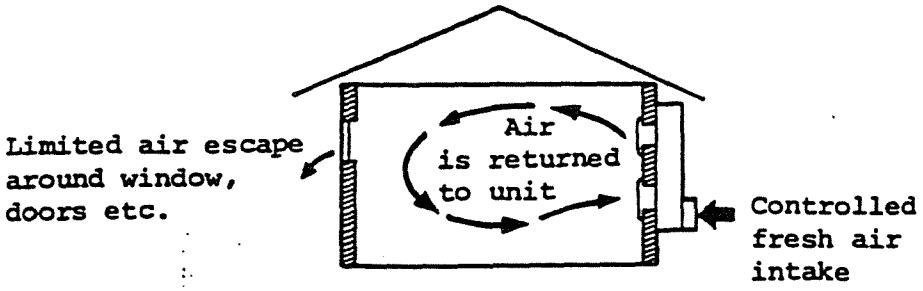
Typical evaporative cooler.

Evaporative air conditioning is cheaper than mechanical air conditioning but requires acceptance by the user that:

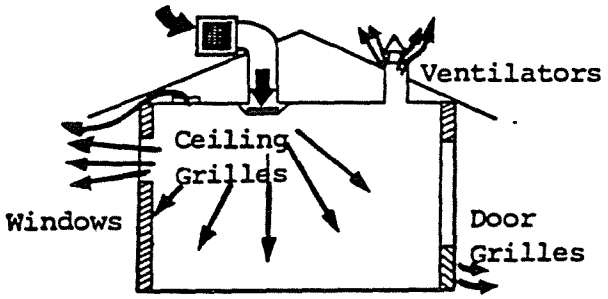
a) The temperature achieved depends upon the outside air humidity rather than temperature.

b) All air brought into the space by the unit must be exhausted at the same rate.

c) The higher air volume needed may result in higher noise levels and draughts. Figure 19.21 shows relative requirements of 'conditioned' and evaporative cooler air requirements.



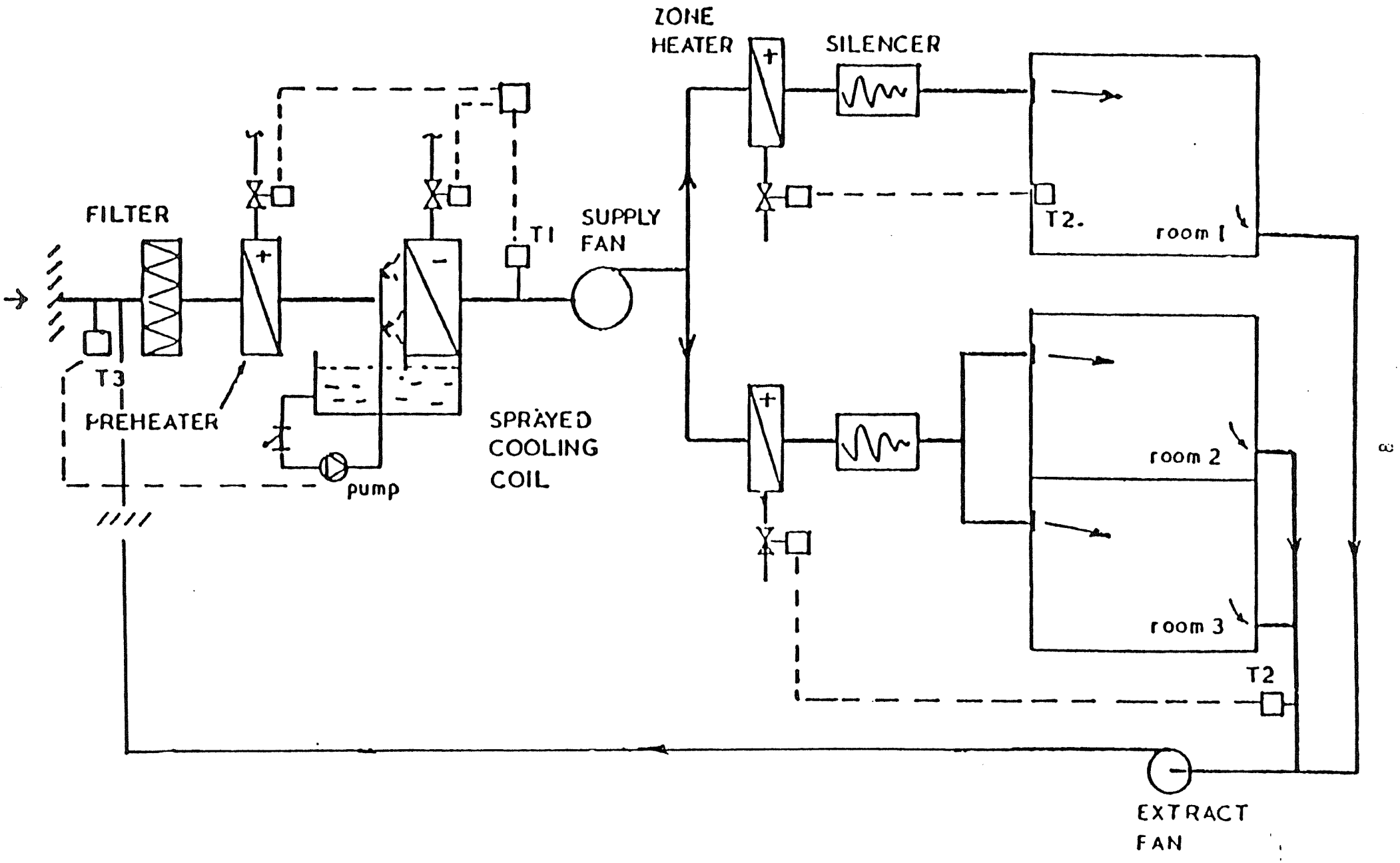
(a) Air conditioners need a closed system for recirculation.



(b) Evaporative coolers need large open areas for air release.

Fig. 19.21

Evaporative cooler requirements.


ZONE
HEATER
SILENCER
FILTER
T1
SUPPLY
FAN
T2.
room 1
T3
PREHEATER
SPRAYED
COOLING
COIL
pump
room 2
room 3
T2
EXTRACT
FAN

VENTILATION NR13

LEARNING OUTCOME 1

STUDENT WORKSHEET 1:1

Question 1.
Name the two most common types of ventilation in use today.

Question 2.
Why do we require proper ventilation?

Question 3.
What is the major disadvantage of natural ventilation?

Question 4.
How was the first attempt at mechanical ventilation achieved and what was it called ?

Question 5.
What is meant by the term induced ventilation?

Question 6.
How is forced ventilation achieve?

_____

_____

_____

_____

Question 7.
List four (4) applications where ventilation is required and the type of ventilation.

_____

_____

_____

_____

_____

Question 8.
There are six (6) considerations to be aware of when using ventilation,name four (4).

_____

_____

_____

_____

Question 9.
What is the difference between ventilation fans and circulation fans?

_____

_____

_____

_____

Question 10.
What does the term "climate- controlled conditions" relate to and what does it apply to?

_____

_____

_____

_____

_____

Question 11.
What are the five (5) factors to be controlled in the air conditioning process?

_____

_____

_____

_____

_____

_____

Question 11.
What do you need to control to obtain desirable indoor conditions?

_____

_____

_____

_____

Question 12.
What air velocities are required for:

	HEATING	COOLING
Office workers:	_____	_____
Restaurants :	_____	_____
Dancing :	_____	_____

Question 13.
Comfort depends upon WHAT conditions?

Question 14.
With the aid of a drawing describe three different duct layout systems.

(A)

(B)

(C)

Question 15.
What is an abrupt expansion and how can some of the resistance created by them be overcome?

Question 16.
Describe spot cooling and give an application.

_____

_____

_____

_____

Question 17.
What regulates the amount of cooling that can be obtained from the use of Evaporative cooling?

_____

_____

_____

Question 18.
Although Evaporative cooling is cheaper to run,what other requirements of Evaporative coolers?

(A)_____

(B)_____

(C)_____

LEARNING OUTCOME 3

Assessment: Practical exercise, student report.

Performance:

a. State the application of several different ductwork components.

b. Select from an installation specification all duct work component items required to install the system.

c. List what installation equipment would be needed to carry out the installation in part b.

FANS

- Classifications and types
- Construction materials
- Applications
- Service requirements
- Operating characteristics
- Identification
- Bearings and shafts
- Power consumption
- Balancing
- Fan laws

Suggested teaching time: 4 hours.

FILTERS

- Impurities in air
- Particle size
- Types of filters
- Applications
- Service/maintenance requirements
- Odour removal
- Absolute filters
- Service schedules
- Problems arising through lack of service
- Correct installation

Suggested teaching time: 4 hours.

Learning outcome 3 (NR13)

FANS

INTRODUCTION

A fan is a pump, having a specific purpose of producing sufficient pressure to overcome system resistance at the required flow rate. A fan pumping air must have a relatively high compressibility of the air.

By proper use of fan manufacturers published catalogue data based on tests, the tables, or preferably curves, make it possible to select a specific fan for a particular application. Tables are most readily available because they are easily obtained from direct computer printout. However, complete curves supply information which tables cannot. Curves can be developed from the tables but the curves will be incomplete since it is common to tabulate only the useable portions of the curves.

In addition to the tables and curves, a set of mathematical relationships exists which make it possible to predict reasonably well the operating conditions other than those of the original selection and installation. These are commonly called the *fan laws*. With these formulae, it is possible to determine required new physical conditions such as speed and horsepower, based on field measurements. The formulae minimise the guess-work which would otherwise be required to change the fan operation in the event of unexpected field conditions. In addition to the presentation of the basic information regarding fans and their operation, the fan laws are presented here with illustrations for their use. Further examples will be presented in the chapter concerning trouble-shooting system malfunctions.

TYPES OF FANS

A fan basically consists of a number of blades on an axle or shaft which together rotate in a housing designed to minimise the internal losses and to maximise the available pressure and air flow at the consumed horsepower. The motor may be directly connected to the fan shaft or may drive the fan through a pulley and belt arrangement which is more flexible. The belt and pulley arrangement is generally more desirable from the standpoint of equipment safety since the belts can break or slip under overload conditions with less chance of damage to motor or fan.

Fans are manufactured in all shapes and sizes. Many are designed for a specific application or piece of equipment. However, fans used in environmental systems fall into two general categories, centrifugal and axial flow.

The centrifugal flow fan is so named since it produces air flow along or in the same direction as the length of the fan shaft, or axially. In both cases, the term axial is derived from the direction of the centreline of the axle or shaft.

Centrifugal fans are classified according to the blades on the impeller frame or wheel.

They are forward curved, backward curved or radial (straight). Modifications of these basic categories produce single width, single inlet (SWSI); double width, double inlet (DWDI); floor mounted; in-line; and tubular centrifugal, among others.

The basic blade configurations and their modifications are indicated in figure 1.2-1.

The backward curved, air foil and backward inclined blades illustrated in figure 1.2-1 are all in the backward curved family, the difference in each case being the blade shape.

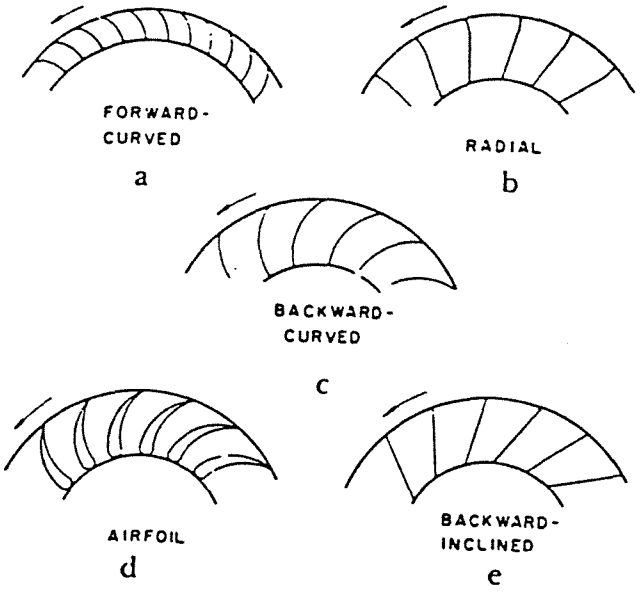


Figure 2-1 Basic Fan Blade Configurations

The SWSI and DWDI fans may be designed with any of the blade arrangements on the wheel. The basic difference in each case is the shape. Each is built with the wheel enclosed in a housing called a scroll which is designed to produce the most efficient airflow within the fan from inlet to outlet (discharge).

The single width fan provides one path for air to flow from the single side inlet, through the wheel and scroll, and out the discharge. Because of the need to provide maximum air distribution across the width of the wheel and blades, increase of size and capacity is accomplished mainly by increasing height since an increase in width would produce an ineffective portion of the blade and wheel on the blind side opposite the inlet. The double inlet fan, with inlets on both sides, provides the means for air to reach this difficult area.

Hence the comparison by name of single and double width. For the same capacity, a single inlet fan is about 30% taller but only about 70% as wide as a double inlet. The DWDI fan can therefore be extremely useful where physical installation height is limited.

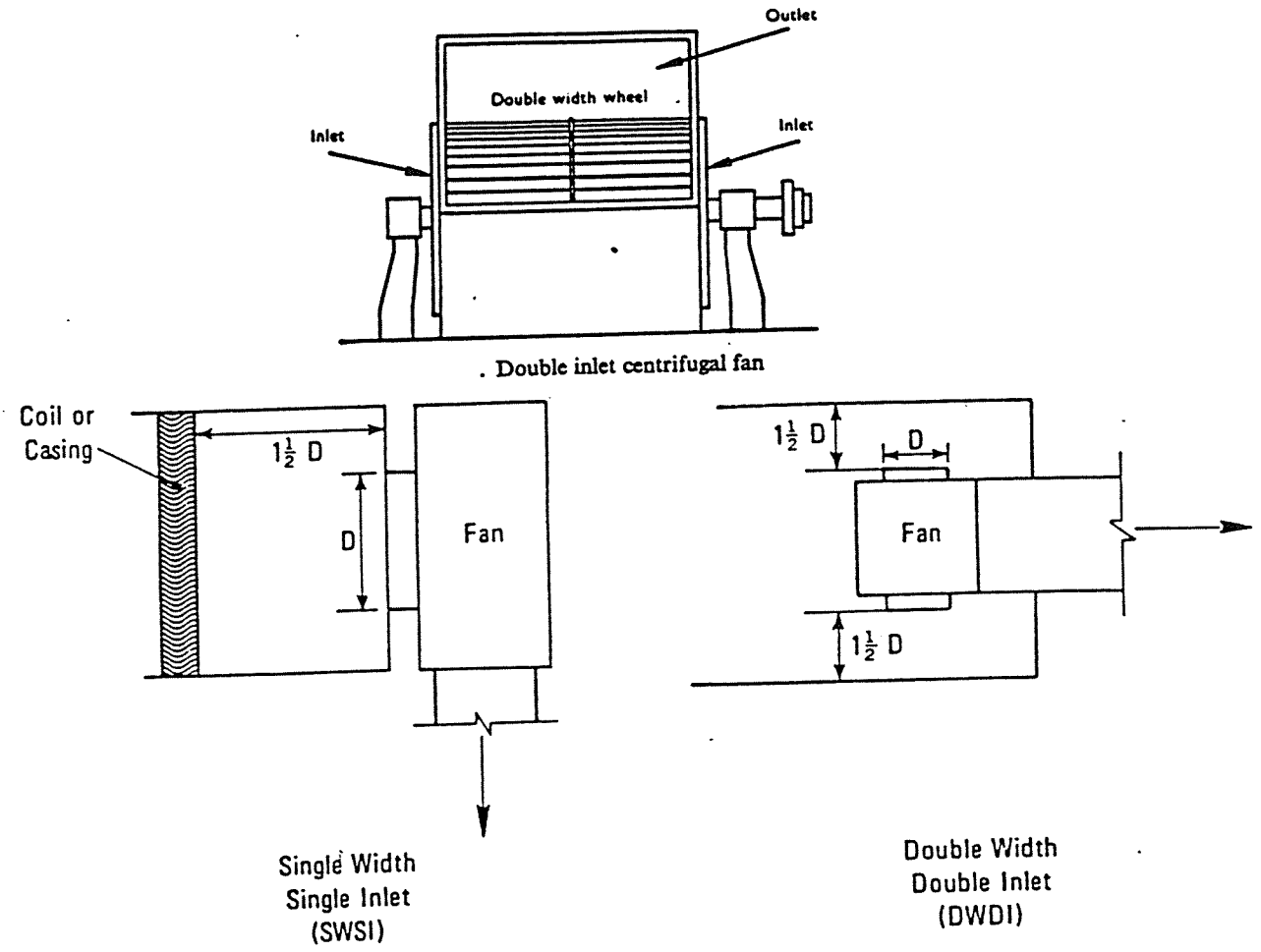


Figure 1-2 Fan Inlet Conditions

In either case, SWSI or DWDI, successful operation of the fan is critically dependent upon the space conditions afforded the air at the fan inlet (see figure 1.2-1). The absolute minimum for proper operation is that the distance from casing to the fan inlet be equal to the entrance opening diameter. Preferably, the distance should be 1½ times this diameter or greater. It can be seen then that the DWDI fan requires a considerable plenum width to provide adequate inlet space for satisfactory operation. Furthermore, the DWDI fan location must usually be in a plenum, where the SWSI fan can be located outside the plenum with the inlet connected to an opening in the side of the casing which must be there to accommodate discharge ductwork connection.

Furthermore, the DWDI fan will commonly have its motor and drive in the air stream which will increase heat gain in the air stream and possibly increase maintenance to belts, pulleys bearings and motor.

The "floor mounted" category referred to does not intend to eliminate the installation of this type of fan on platforms, roofs or elsewhere other than the floor itself. It is actually a comparative location description with respect to the tubular centrifugal which is mounted directly in the ductwork. In this latter instance, the fan housing is tube shaped and the fan is turned so that the air enters axially. After passing through the fan, the air is discharged toward the side of the housing and must be redirected 90° back to the axial direction. The intent is to conserve floor space. It must be recognised that velocities are often high to keep size down, and installation conditions at inlet and outlet, as well as internal design provisions, are extremely important to prevent excessive losses and noise.

It is possible, as in the case of the axial fans, that incorrect installation can practically destroy the effectiveness of the fan by excessive local pressure losses.

CENTRIFUGAL FANS

The centrifugal fan comprises an impeller which rotates in a casing shaped like a scroll as illustrated in figure. 2-1. The impeller has a number of blades or plates around its periphery, similar to a water wheel or the paddle wheel of some shallow draught river steamers. The casing has an inlet on the axis of the wheel and an outlet at right-angle to it. When the impeller rotates the blades at its periphery throw off air centrifugally in a direction following the rotation. The air thrown off into the scroll is forced out of the outlet as more and more leaves the blades. At the same time air is sucked into the inlet to replace that which is discharged radially. The purpose of the scroll is to covert the high velocity pressure at the blade tips into static pressure.

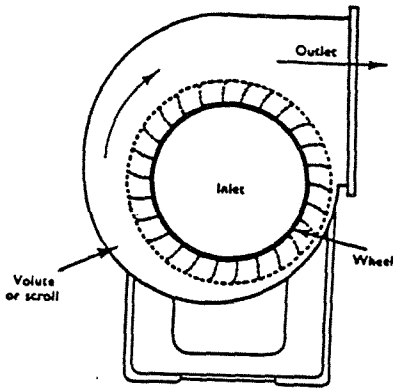
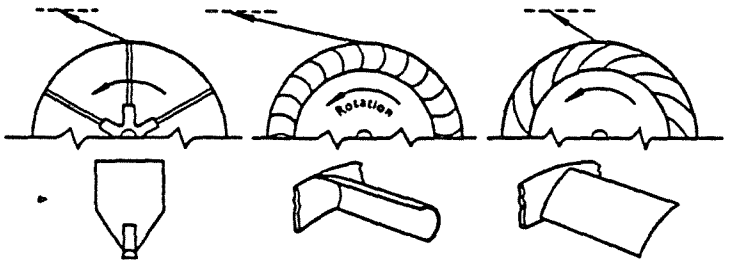


Fig. 2-1. General arrangement of centrifugal fan

The three different types of blades used are shown in Figure 2.2



. Types of centrifugal fan blades

The shape of the blades influences for force exerted on the air and the proportion of energy imparted in the form of velocity. The velocity of air leaving the impeller is proportional to the length of the arrows in figure 2.2.

The efficiency of centrifugal fans suffers from the fact that the air handled must turn through 90°. This causes losses of energy due to shock and eddies. Moreover the aerodynamic efficiency of the scroll, or volute as it is called, is generally rather low. The fan efficiencies are usually between 45% and 75% according to type.

APPLICATIONS

The centrifugal fan is used in most comfort applications because of its wide range of quiet, efficient operation at comparatively high pressures. In addition, the centrifugal fan inlet can be readily attached to an apparatus of large cross-section while the discharge is easily connected to relatively small ducts. Air flow can be varied to match air

distribution system requirements by simple adjustments to the fan drive or control devices.

FORWARD CURVE FANS

Higher efficiencies are obtained when the blades have curved surfaces. A common form of curved blade has the concave side facing the direction of rotation. Blades of this type are shallow as "multi-vane" or "multi-blade". The blades are mounted between side rings mounted on the arms of a spider or on a solid plate mounted on the shaft. A typical design is shown in figure 3.1. The forward-curve blade has a scoop effect on the air. As shown in figure 13-b, the velocity of air leaving this type is greater than with others. As a result this design moves more air than others for a given impeller diameter and speed. In other words, for a given capacity the forward-curve fan is smaller and runs at a lower speed.

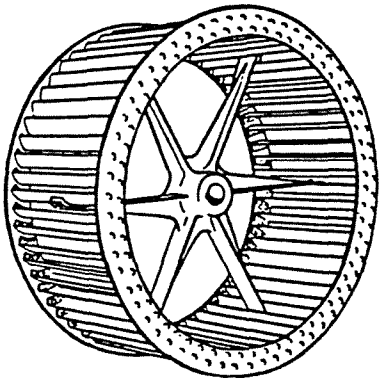


Figure 3-1. Forward curve impeller

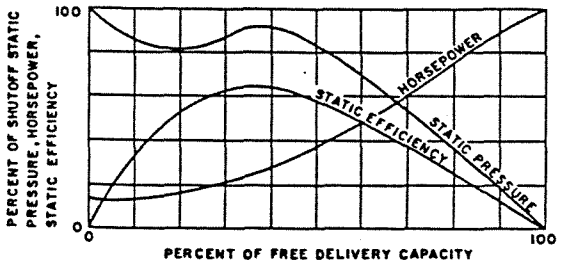


Figure 3-2. Forward Curved Blade Fan Performance.

The variation of volume with pressure for forward-curve fans. The rise of horse-power towards maximum volume is even more marked with this type paddle-blades. This factory considerably affects the rating of the fan drive required.

ADVANTAGES:

1. Runs at relatively low speed compared to other types for same capacity.
2. Smaller fan for given duty, excellent for fan-coil units.

BACKWARD-CURVE FANS:

The highest efficiencies in centrifugal fans are achieved with backward-curve blades. These have the convex side facing the direction of rotation. This form improves air flow through the blades by reducing shock and eddy losses. These fans operate at higher speeds than other types of centrifugal fans. The blades are longer radially than the forward-curve type and usually heavier while the impellers are strongly reinforced with stiffening rings and larger section shafts are required.

The air output for a given wheel diameter is less than with forward-curve fans. The efficiency, however, can be substantially higher. Special types may develop very high pressures, for example, forced draught fans for boilers.

The pressure/volume characteristics of backward-curve fans are shown in figure 4-1. In this case the maximum horse-power occurs within the normal working range.

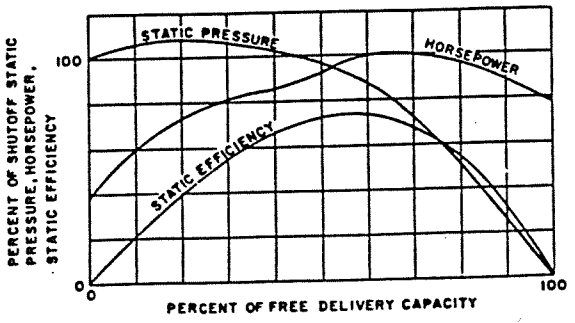


Figure 4-1. Backward-curved blade fan performance

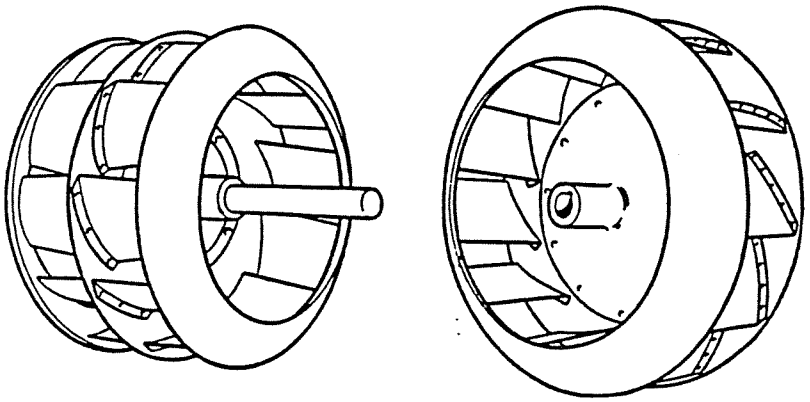


Figure 4-2. Backward-curve impellers

Two modifications of the backward-curved blade fan are the air-foil and backward-inclined blade fans.

These are illustrated in Figure d and e. Both are non overloading types.

The airfoil blade fan is a high efficiency fan because its aerodynamically shaped blades permit smoother air flow through the wheel. It is normally used for high capacity, high pressure applications where power savings may outweigh its higher cost. Since the efficiency characteristic of an airfoil blade fan usually peaks more sharply than those of other types, greater care is required in its selection and application to a particular duty.

The backward-inclined blade fan must be selected closer to free delivery; therefore, it does not have as great a range of high efficiency operation as does the backward-curved blade fan. Manufacture of an inclined blade is understandably a simpler operation.

ADVANTAGES:

1. More efficient.

2. Horsepower curve has a flat peak so that the motor may be sized to cover the complete range of operation from zero to 100% air flow for a single speed. Non overloading.

3. Pressure curve is generally steeper than that of the forward-curved fan. This results in a smaller change in air volume for any variation in system pressure for selections at comparable percentages of free delivery.

4. Point of maximum efficiency is to the right of the pressure peak, allowing efficient fan selection with a built-in pressure reserve.

5. Quieter than other types.

RADIAL BLADE FAN

The radial blade fan has efficiency, speed and capacity characteristics that are midway between the forward-curved and backward-curved blade fans. It is seldom used in air conditioning application because it lacks an optimum characteristic.

Typical performance of a radial (straight) blade fan is shown in Figure 5-1. The pressure characteristic is continuous at all capacities. Horsepower rises with increasing air quantity in an almost directly proportional relation. Thus, with this type of fan the motor may be overloaded as free air delivery is approached.

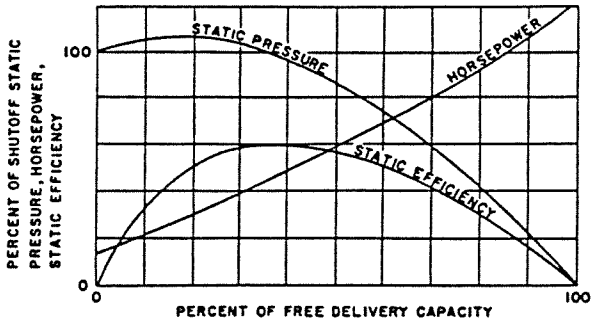


Figure 3-1 Radial Blade Fan Performance

ADVANTAGES:

1. Self-cleaning.

2. Can be designed for high structural strength to achieve high speeds and pressures.

AXIAL FLOW FANS

Axial flow fans are classified as propeller, tube axial and vanaxial (see figure 6-1). All may be belt driven or directly connected to the motor.

The most commonly recognised is the propeller fan, which may be a small household version, a pedestal model, a kitchen wall, exhaust fan or one of the many other adoptions available. This style is also the least expensive to produce. The blades are often stamped metal and in many cases little consideration is given to desirable features, such as entering air pattern and tip shrouds which can greatly improve performance. It may be seen from examination of the curves later in this chapter that the static pressure characteristics of this type of fan are comparatively poor which leads to the correct conclusion that restrictions such as ductwork, louvers, screens, and dampers have a serious effect on performance. In fact it is not generally good practice to apply this type of fan where ductwork is required and other restrictions should be kept to an absolute minimum. Some fan designs provide ratings at low external resistance figures, but it is still prudent to be generous in the original selection. The propeller fan is also a relatively low speed fan. Increased capacity means increased fan diameter and resulting high blade tip speed which also limit the capacity performance.

Vaneaxial and tubeaxial fans are specially designed propeller types fans, although their operating characteristics with external resistance are greatly improved. The basic difference between the two may include blade configuration but is mainly the fact that tubeaxial fans do not employ guide vanes before or after the impeller and the vaneaxial fans do. The vanes are arranged in such a way as to improve the internal air flow pattern, reduce the internal resistance and thus to increase the efficiency and performance of the fan. The blade and vane configuration may be flat or curved, single or double thick, the selection being determined in the design to produce best performance at lowest cost.

Two critical items must be considered in the selection and application of axial fans.

One is similar to the tubular centrifugal. The outlet and especially the inlet air flow pattern is extreme importance. Some axial fans may be applied without inlet and outlet cones, but this refinement usually improves operation and in many cases is the difference between success and failure. The "streamlining" effect of the cones is similar to that of a venturi meter, where the axial fan casing is the centuri and the fan is placed at the narrowest point in the air path. The second major consideration is that of sound. The axial tube fan can be selected to produce high volumes of air in a relatively small piece of equipment, but the noise level becomes high. In locations where background noise is already high or unimportant, this may not be serious consideration.

However, when noise is a factor, special care must be taken in the fan selection. In many cases sound attenuators either ahead or after the fan or both may become an absolute must.

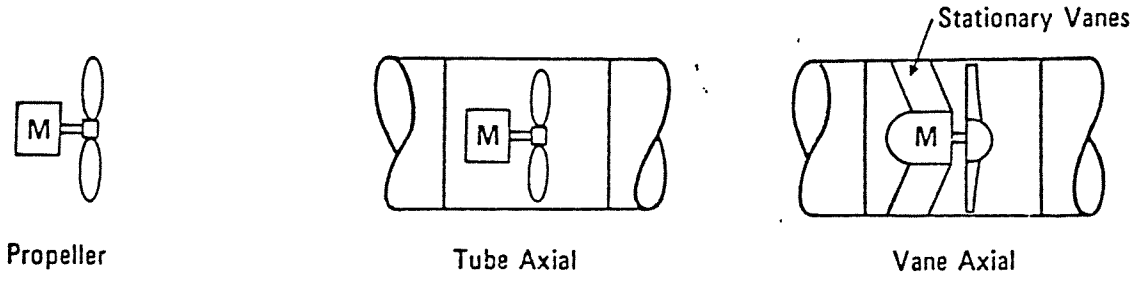


Figure 6-1 Axial Fan Diagrams

AXIAL FLOW FANS

The tubeaxial fan is a common axial flow fan in a tubular housing but without inlet or outlet guide vanes. The blade shape may be flat or curved, of single or double thickness.

The axial flow fan has become particularly associated with the vaneaxial type which has guide vanes before or after the fan wheel. To make more effective use of the guide vanes, the fan wheel usually has curved blades of single or double thickness. Figure 6:1-1 is a sectional view of the vaneaxial fan.

The curved stationary diffuser vanes are the type most frequently used when higher efficiency vaneaxial fans are desired. The purpose of these vanes is to recover a portion of the energy of the tangentially accelerated air.

Typical performance of an axial flow fan is shown in Figure 6:1-2.

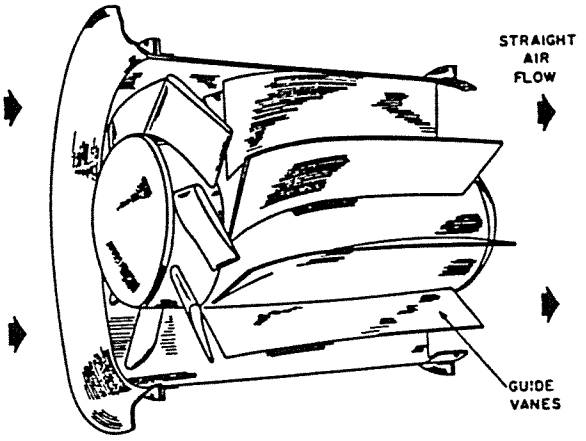


Figure 6:1-1 Vaneaxial Fan

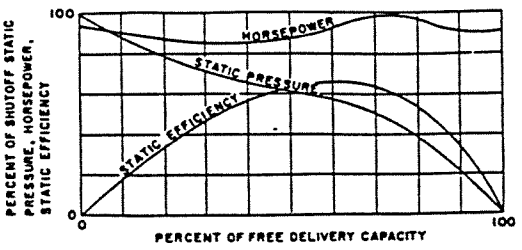


Figure 6:1-2 Axial Flow Fan Performance

PROPELLER FANS

Propeller fans serve a very wide field of applications where resistance to air flow is low. As a rule they are used where there is no duct system or where the length of duct is short. In a majority of cases they move air through a hole in a wall. Their inherent attribute is that they enable large volumes of air to be handled economically and entail low capital costs. Consequently they are extensively used all over the work for general ventilation purposes.

The term exhaust fan is often applied to propeller fans because they are so widely used for exhausting air from buildings. That description, however, is not a correct one, since the fans may be used for fresh air input and for many other purposes, such as for unit heaters, coolers, etc.

These fans have an impeller with two or more blades, usually of sheet steel, set at an angle to the hub, somewhat in the manner of a ship's propeller. In some cases the blades follow the shape of aircraft propellers, but sheet steel blades of the type shown in Figure 6:2-1 are widely used.

The propulsive effect of the blades varies according to their shape. A fan with broad curved blades will move more air and is quieter than one with flat and narrow blades of the same diameter running at equal speed.

Air enters a propeller fan from all directions and is discharged mainly axially, but there is also some radical discharge. When resistance is imposed the air tends to flow back through the impeller. Because of this propeller fans are not suitable for working against any substantial resistance. Their particular field is air movement under condition of free intake and discharge, or at static pressures not exceeding 0.5 in wg. For applications of this kind they are usually the most practical and most economical choice.

The power loading of a propeller fan increases with the resistance against which it works. When working against excessive resistance the power input may rise very considerably until the motor over-heats and eventually burns out.

As a safeguard the motors should be generously rated.

As a general rule a propeller fan with sheet steel blades gives maximum volume under free air flow conditions when the straight training edge of the blades, that is, the discharge edge, is flush with the mounting ring as shown in Figure 6:2-1. Maximum pressure is obtained by projection permits centrifugal discharge from the blade tips, reduces losses through backflow, and thereby enables the fan to develop its maximum pressure.

Ample space should be allowed at the blade tips for centrifugal discharge. If a fan is mounted in a circular duct the best performance will be obtained when the duct is not less than 25% larger than the impeller diameter and the fan is mounted on a diaphragm plate with the blades projected through the orifice.

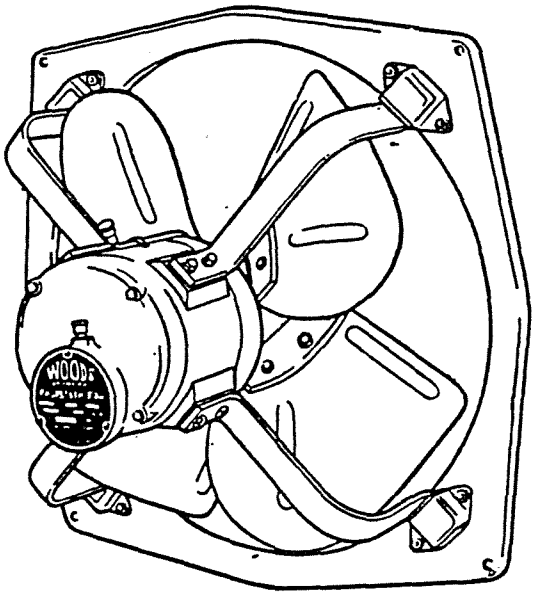


Figure 2-1 Ring-mounted Propeller Fan

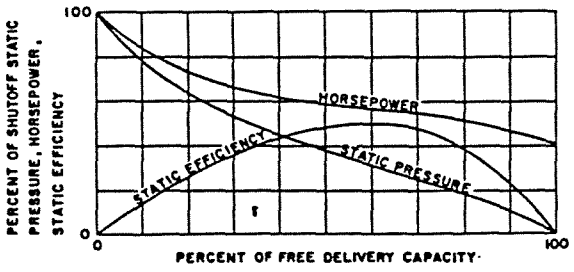


Figure 2-2 Propeller Fan Performance

FAN CONSTRUCTION

In addition to the physical variations required by the types of fans themselves, there are requirements imposed upon the construction as a result of system pressures. The Air Moving and Conditioning Association (AMCA) has established construction classification standards for the industry which, though not mandatory, are followed by most centrifugal fan manufacturers.

Formerly, the operating limits were specified by the total pressure which a fan must be able to develop on at least one point of its performance curve for a given class of construction. The performance limits were then determined by the maximum impeller speed for that construction and then catalogued using this criteria. The requirements were:

Class I	95.2mm total pressure
Class II	171.4mm total pressure
Class III	311mm total pressure
Class IV	Aobe 311mm total pressure.

This method of classifying construction was a source of much confusion in the industry. Later AMCA standards define the minimum outlet velocity versus static pressure limits necessary for Class I, II, and III construction. The curves in figure 7:1 illustrate these

requirements for single width, backward curved blade fans. Similar information for other fan types may be obtained by referring to the AMCA standards. The classifications are based on a "mean brake horsepower per square metre of outlet area" concept and more nearly represent the actual structural limitations of the centrifugal fan designs. A centrifugal fan in one of these classifications must be physically capable of performing over the entire class range.

From the user's point of view, the limit lines represent minimum performance. For example, the Class II fan in the chart must now be capable of providing 216mm of static pressure at 15.24 metres per second outlet velocity. By comparison, the old standard merely required that the Class II fan be capable of operation up to 171mm of total pressure at one point on its performance curve.

Another AMCA classification is especially for cabinet fans. These are: Class A, to 76mm wg; Class B, to 139 wg; Class C, above 139mm wg.

FAN INSTALLATION/MOUNTING

Supporting a fan is not merely a matter of strategically placed steel and concrete. The fan, motor, and drive together became a complex system of dynamic components, all connected together by moving in different directions at different relative speeds. The composite assembly must be tamed by the proper application of supports and isolation.

Isolation of the fan, motor, and drive from the connecting ductwork is generally accomplished by means of a suitable flexible connection. The connection must be long enough so that the rigid duct and the moving equipment do not touch. The differences between the position in motion must be taken into account when the initial installation is made. Even the difference in position caused by the static loading of the fan isolators must be accounted for. Too often the installation is made from a straight line drawing which does not account for the variations of misalignment, torque and statice and dynamic deflection of bases and isolators. Then the duct and equipment contact, negating the purpose of the flexible connection and transmitting vibration up and down stream of the equipment.

Even though the direct equipment vibration may be isolated by the flexible connection in the duct, the air in the system, because of velocities and pressures or because of fittings, may generate motion in the ducts. At times it is necessary to use vibration isolators on the duct hangers to overcome these disturbances.

Motion of the fan, motor and drive mass may be reduced in magnitude or amplitude by adding weight to the system. A common example is the use of a concrete and steel base. At other times, the mass is not added, and the isolators are selected to operate at the fan and motor system amplitude. In almost every case, isolation is required, and the type will depend on the application. The means of isolation selected must isolate the vibration at the generated frequencies and most also adequately support the system. The isolators must never be allowed to over-compress or "bottom out" so that they short circuit. They must never be so flexible that the system will warp and cause misalignment and unexpected, unreasonable wear to belts, bearings and other components. In some instances where the building construction is light or where a resonant condition might

amplify the vibration, the isolation system must protect against what may be called sympathetic resonance in the construction. At times the result can be worse than having no isolators at all.

The four common types of isolators are steel coil springs, double rubber in shear, single rubber in shear, and cork, in order of decreasing efficiency and cost. These materials are often used alone, in combination with each other and in various types of load distributing and supporting rails, frames, and bases. For further information, refer to the later chapter regarding this subject.

By proper use of vibration isolation equipment, a large portion of the noise normally generated by fans and other equipment may be prevented or reduced to acceptable levels. However, there are other sources of undesirable sound. Fans generate noise in the process of pumping air, and air generate noise from velocity and turbulence while moving through the system. Any fitting, outlet, damper, air valve, straight piece of duct, or other sources of undesirable sound. Fans generate noise in the process of pumping air, and air generates noise from velocity and turbulence while moving through the system. Any fitting, outlet, damper, air valve, straight piece of duct, or other system device, including sound attenuators (deadeners) and traps may generate noise under some conditions.

Fans generate sound in a relatively narrow and predictable range of frequencies dependent upon the rotational speed and the number of individual impeller blades.

CONCLUSION

The following information concerning the characteristics of the various fan types will assist in determining the fan selection, providing the previous considerations are satisfied or are not determining factors.

CENTRIFUGAL

Forward curved blade fan - As system static pressure falls off, the horsepower increases immediately, and continues to increase. For the illustration the motor would begin to overload immediately, although small increases may remain within the service factor.

Backward curved blade fan - Horsepower has been selected at the maximum requirement at any point on the horsepower curve, and neither increasing nor decreasing system static pressure will increase horsepower requirement or cause motor overload.

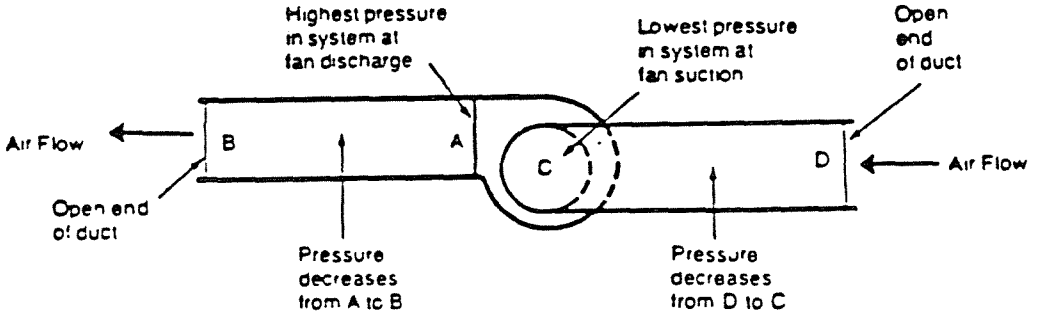
Radial blade fan - Operation is similar to the forward curved blade fan with the same results.

Propeller fan - Horsepower requirements are relatively constant throughout the range but decreasing system pressure could cause overload if the motor is initially selected too close to name-plate rating or within the service factor.

PRINCIPLES OF AIR FLOW

For large air conditioning systems and package air conditioners which serve several zones, the treated air must be conveyed through ducts.

Air flows through a duct because the pressure existing at one point is higher or lower than the pressure at another point of the same duct. The direction of pressure increase of decrease decides the direction of air flow, ie that is towards the lower pressure.



Fan creates a difference in pressure, causing air to flow in duct

The difference in pressure is created by the fan. The air does not normally move along in a placid stream; it moves in either turbulent or laminar flow.

LAMINAR OR TURBULENT FLOW

When air flows through a duct at low velocities the particles follow paths free from eddy currents of swirls. The flow is then said to be *LAMINAR.* As the velocity increases, the characteristics of flow change; eddy currents form and the air becomes swirly. This type of flow is known as *TURBULENT* flow. While laminar flow develops less resistance than turbulent flow it can sometimes cause problems of stratification.

The *volume of air* handled by a fan is the flow rate produced by a fan independent of the density of the air normally expressed as:

Cubit metres per second handled by a fan at any density
$m^3/_s$
or Litres per second handled by a fan at any density.
l/s

OUTLET VELOCITY

Outlet velocity is the theoretical velocity of air as it leaves the fan outlet, and is calculated by dividing the air volume in m^3/s by the fan outlet area in m^2.

Because of the variations in velocity across the fan outlet, velocity readings taken across a fan outlet mean very little. Readings ~~should be~~ taken across a fan outlet mean very little. Readings should be taken further down stream of the discharge duct to allow air flow to become reasonably uniform.

Outlet velocity is normally associated with centrifugal fan and *Tip Speed* is often quoted for axial fans, although both terms can be applied to either form of fan design.

SYSTEM PRESSURES

a. Static Pressure (P_s) is the pressure within a duct which tends to burst the duct.

b. Velocity Pressure (P_v) is the pressure which air in a duct exerts due to its motion.

c. Total Pressure (P_T) is the sum of the static pressure (P_s) and the velocity pressure (P_v).

$$P_T = P_S + P_V$$

PRESSURE CLASSIFICATION OF DUCTS

Classification of duct systems by pressure and/or velocity are quite arbitrary. The *SHEET METAL* and *AIR CONDITIONING CONTRACTORS NATIONAL ASSOCIATION, INC. (SMACNA)*, in developing duct, construction standards, established the following breakdown:

a. **Low velocity**

Up to 10.0 metres/second and up to 50mm (500 Pa) swg static pressure.

b. **High velocity**

Above 10 metres/second.

i. Medium Pressure: Up to 150mm (1500 Pa) swg static pressure.
ii. High Pressure: Over 150mm (1500 Pa) swg static pressure up to 250mm (2500 Pa) swg static pressure.

Note: Medium and high pressure ducts should be tested for leaks because performance of systems having such ductwork is affected by air leaks to a greater extent than low pressure systems.

In parts 18, 19 and 20, we covered the definitions of:

a) volume of air handled;

b) outlet velocity;

c) total, static and velocity pressures.

We are now going to look at *Static Pressure (Ps)* as it effects fan performance. Consider a complete air distribution system which may include heat exchangers, (coils) filters, grilles, dampers and duct work. If air is forced through the system at a given rate of flow, a certain static pressure will result. If air is forced through the same system at a

different flow rate, a different static pressure will exist. Most air distribution systems are turbulent flow systems in which the static pressure varies as the square of the air flow rate.

$$Ps = A\ Q^2$$

where Ps = STATIC PRESSURE (Pa)
A = CONSTANT
Q = VOLUME FLOW RATE (m^3/s)

The graph constructed from this equation is called:

THE SYSTEM CHARACTERISTIC CURVE.

Field problems often will required that both a fan performance curve and a system curve be used. If a fan curve is available, a system curve can be drawn directly on it. Where manufacturers data is in a multi rating table form it is possible to draw a fan curve based on these tables.

EXAMPLE:

A test on a forward curved centrifugal fan gave the following results with the fan running at 480 RPM.

Volume (m^3/s)	0	1.0	2	3	4	5	6	7
Pressure (Pa)	750	675	650	700	675	550	350	125

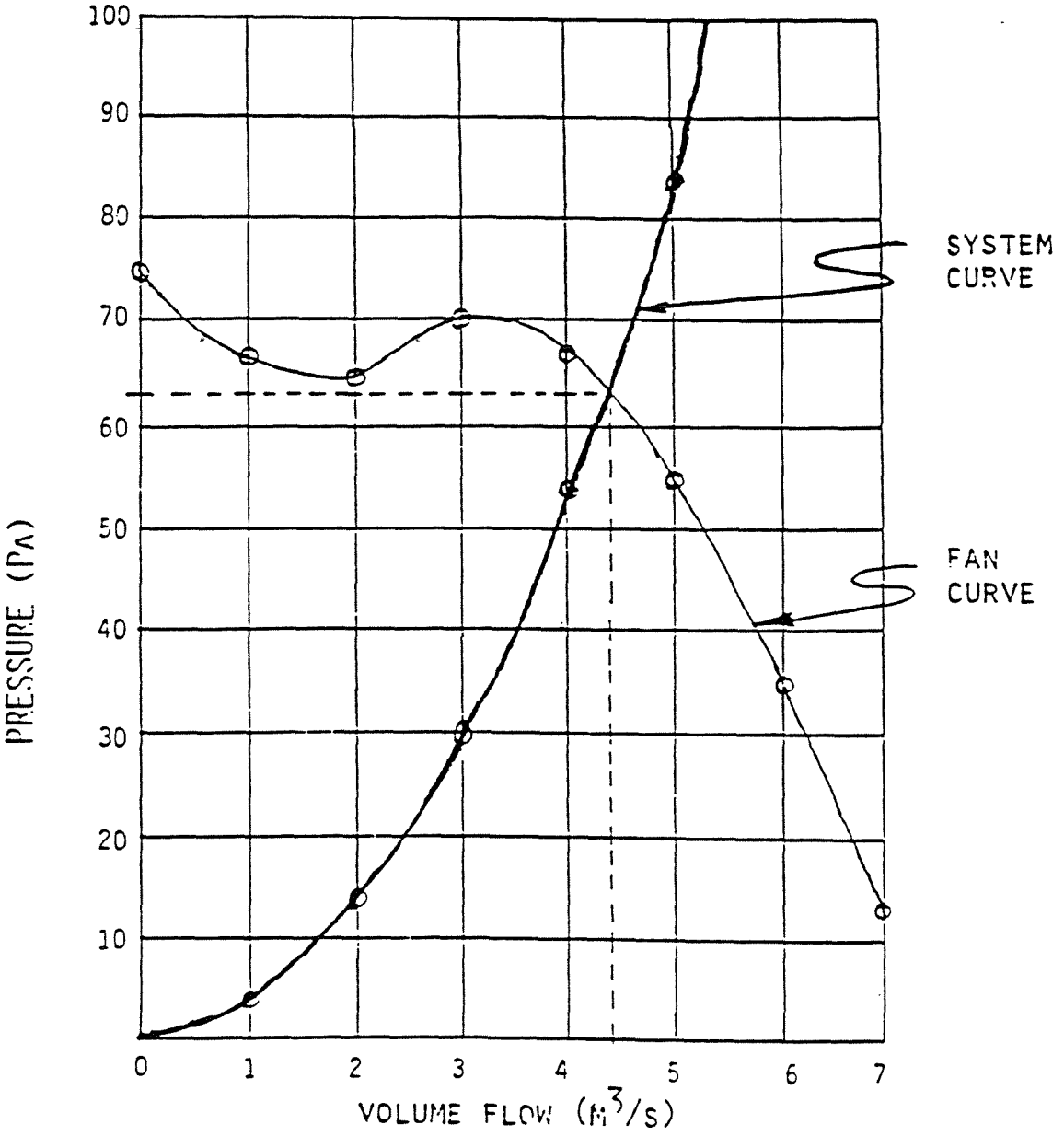
The fan supplies air to a system for which the static pressure loss is calculated to be 615 Pa when the volume flow rate is 4.3 m^3/s. Using the graph below, plot the fan and system curves.

SOLUTION

$$Ps = A\ Q^2$$
$$615 = A\ 4.3^2$$
$$A = \frac{615}{(4.3)^2} = 33.3$$

For selected volume flow rates, find the corresponding static pressure losses.

Volume (m^3/s)	1.0	2.0	3.0	4.0	5.0
Pressure (Pa)	33.3	133.2	299.7	532.8	832.5



The fan characteristics (volume flow rate versus static pressure) and system characteristics are now drawn on the graph. The condition at which the fan operates is given by the point at which the two characteristics intersect.

Approximate Volume	=	4.4 m^3/s
Approximate Static Pressure	=	630 Pa

When the fan is actually tested on start-up, it is found that the actual flow rate is only 2.25 m^3/s and the measured static pressure is 557 Pa. The obvious solution is to increase the fan speed, however, if the fan is operating in the unstable or *surge* area, the static pressure must be reduced or another fan substituted.

To find the actual system characteristic:

$$Ps = A\,Q^2$$

$$\therefore \quad A = \frac{557}{(2.25)^2} = 110$$

Volume (m^3/s)	1.0	1.5	2.0	2.5	3.0	3.5
Pressure (Pa)	110	247.5	440	687.5	990	1347.5

Plot the actual system capacity on the graph and you can see that the static pressure cuts the fan curve in more than one place. Therefore, the fan may settle down to any flow rate where static pressure cuts the fan curve or it may surge between them.

FAN LAWS

It is often necessary to operate a fan at a speed other than the manufacturers tabulated capacities due to the actual required volume of air.

The following is confined to those laws of most common use in air conditioning.

For a given Fan Size, Duct System and Air density.

i. The volume varies proportional to the fan speed:

$$\frac{Q1}{Q2} = \frac{N1}{N2}$$

ii. The developed pressure is proportional to the square of the fan speed:

$$\frac{P1}{P2} = \left(\frac{N1}{N2}\right)^2$$

iii. The power absorbed is proportional to the cube of the fan speed:

$$\frac{W1}{W2} = \left(\frac{N1}{N2}\right)^3$$

where:
Q = Volume flow rate (m^3/s)
P = Pressure (Pa) (normally static, but it can be velocity or total pressure)
W = Watts (not installed but what is absorbed)
N = Revolutions per minute (Revs/s)

EXAMPLE

A fan delivers 7.3 m^3/s of air against a static pressure of 383 Pa when rotating at 600 RPM (10 Rev/s) and absorbs 3.9 kW. If the fan speed is decreased to 400 RPM (6.7 Rev/s), find:

a. the new volume;
b. the new static pressure;
c. the new absorbed power.

SOLUTION

$$\text{Volume } (m^3/s) = \frac{Q1}{Q2} = \frac{N1}{N2}$$

$$\therefore Q2 = Q1 \left(\frac{N2}{N1}\right)$$

$$Q2 = 7.3 \left(\frac{400}{600}\right)$$

$$Q2 = \underline{4.87}\ m^3/s$$

$$\text{Static Pressure (Pa)} = \frac{P1}{P2} = \left(\frac{N1}{N2}\right)^2$$

$$\therefore P2 = P1 \left(\frac{N1}{N2}\right)^2$$

$$P2 = 383 \left(\frac{400}{600}\right)^2$$

$$P2 = \underline{170.2}\ Pa$$

$$\text{Power absorbed (kW)} = \frac{W1}{W2} = \left(\frac{N1}{N2}\right)^3$$

$$\therefore W2 = W1 \left(\frac{N2}{N1}\right)^3$$

$$W2 = \frac{3.9}{383} \left(\frac{400}{600}\right)^3$$

$$W2 = \underline{1.16}\ kW$$

PROBLEMS

A fan running at 350 RPM (5.83 Rev/s) supplies 6.2 m^3/s when the static pressure is 187 Pa and absorbs 2.25 kW.

If the speed of the fan is increased to 480 RPM (8 Rev/s), find:

i. the new air volume;
ii. the new static pressure;
iii. the new absorbed power.

A fan is required to supply 700 l/s. When it is started, it is found to have a capacity of 572 l/s when rotating at 400 RPM (6.7 Rev/s) and a static pressure of 478 Pa. The drive motor installed is rated at 1.5 kW and the absorbed power is 850 Watts.

Find:

i. the required RPM;
ii. the new static pressure;
iii. the required absorbed power.
iv. Would the existing motor still be suitable when the required air flow is achieved?

INSPECTION AND PROCEDURES

Before checking the fan and during inspection, the fan must be electrically isolated and all disconnected switches locked in the "*OFF*" position. Where these are remote from the fan prominent "*DO NOT START*" signs should be installed.

SYSTEM CHECKLIST

a. A impeller comes to rest check if rotation is correct.

b. See impeller is not installed backwards.

NOTE: Fan rotation for centrifugal fans is *clockwise* or *counter clockwise* when viewing the *drive side* and for axial fans when viewing the *inlet.*

c. For belt driven fans check alignment of motor and fan pulleys.

d. Check belt tension to the manufacturers recommendations - excessive tension will shorten fan and motor bearing life. *Belt should be re-adjusted after the first 48 hours of operation.* Check condition of pulleys and belts.

e. Check passages between inlets impeller blades and housing for damage from corrosion or foreign matter trapped in impeller.

f. Check ductwork for loose insulation, sheet metal, paper, etc,.

g. Check conditions of coils, filters, etc. If dirty, clean.

h. On fan equipped with inlet vane or damper control, check visually that the vane/damper position agrees with the position of the control arm.

For double width fans check that both inlet/vane/dampers are synchronised. With unbalance flow between inlets, the thrust on the bearings is also unbalanced and it often leads also to surge conditions in the fan.

i. After completing the system check list, put the fan back into operation.

j. Inspect the entire system including the fan, fan plenum and all ductwork for leaks. Leaks may be detected by sound, smoke, feel, soap solution, etc. Common leak sources are access doors, coils, duct seams, fan outlet connection, etc., which must be sealed.

TROUBLESHOOTING CHART

NOISE

a. Impeller Hitting Inlet Ring

PROBABLE CAUSE

i Impeller not centred in inlet ring.
ii Inlet ring damaged.
iii Crooked or damaged impeller.
iv Shaft loose on bearings.
v Impeller loose on bearings.
vi Bearing loose on bearing support.

b. Drive

PROBABLE CAUSE

i Pulley not tight on shaft (motor and/or fan).
ii Belts too loose.
iii Belts too tight.
iv Belts wrong cross section.
v Belts not "matched" in length on multi belt drive.
vi Variable pitch pulleys not adjusted so each groove has same pitch diameter.
vii Misaligned pulleys.
viii Worn belts.
ix Motor, motor base or fan not securely anchored.
x Belts oily or dirty.
xi Incorrect drive selection.

c. Bearing

PROBABLE CAUSE

i Defective bearing.
ii Needs lubrication.
iii Loose on bearing support.
iv Seals misaligned.
v Worn bearing.
vi Corrosion between inner race and shaft.

d. Shaft Seal Squeal

PROBABLE CAUSE

i Needs lubrication.
ii Misaligned.

e. Impeller

PROBABLE CAUSE

i Loose on shaft.
ii Unbalanced.

f. Shaft

PROBABLE CAUSE

i Bent.
ii If more than two bearings are on a shaft, they must be properly aligned.

g. High Air Velocity

PROBABLE CAUSE

i Ductwork too small for application.
ii Fan selection too small for application.
iii Registers or grilles too small for application.
iv Heating or cooling coil with insufficient face area for application.

h. Rattles, Rumbles or Whistles

PROBABLE CAUSE

i Dampers.
ii Registers.
iii Grilles.
iv Sharp elbows.
v Sudden expansion in ductwork.
vi Sudden contraction in ductwork.
vii Turning vanes.
viii Leaks in ductwork.
ix Fins on coils.
x Vibrating ductwork.
xi Vibrating cabinet sections.
xii Vibrating parts not isolated from building.

i. Pulsation or Surge

PROBABLE CAUSE

i Restricted system causing fan to operate at poor point of ratings.
ii Fan too large for application.
iii Ducts vibrate at some frequency as fan pulsations.

LOW VOLUME

PROBABLE CAUSE

i Forward curved impeller installed backwards.
ii Fan running backwards.
iii Fan speed to low.
iv Actual system is more resistance to flow than calculated.
v Dampers closed.
vi Registers closed.
vii Leaks in supply duct.
viii Insulating duct liner loose.

ix Filters dirty or clogged.
x Coils dirty or clogged.
xi Obstructed fan inlet.
xii No straight duct at fan outlet.

HIGH VOLUME

PROBABLE CAUSE

i Oversized ductwork.
ii Access door/s open.
iii Registers or grilles not installed.
iv Dampers set to by-pass coils.
v Filters not in place.

STATIC PRESSURE LOW - VOLUME HIGH

PROBABLE CAUSES

i System has less resistance than calculated. This is a common occurrence. Fan speed may be reduced to obtain desired flow rate. This will also reduce absorbed power.
ii Fan speed to high.

STATIC PRESSURE HIGH - VOLUME LOW

PROBABLE CAUSES

i Obstruction in system.
ii Dirty filters.
iii Dirty coils.
iv System too restricted.

ABSORBED POWER HIGH

PROBABLE CAUSES

i Backward inclined impeller installed backwards.
ii Fan speed to high.
iii Oversized ductwork.
iv Face and by-pass dampers oriented so dampers are open at same time by-pass dampers are open.
v Filters left out.
vi Access door/s open.
viii Fan not operating at efficient point of rating. Fan size or type may not be best for application.

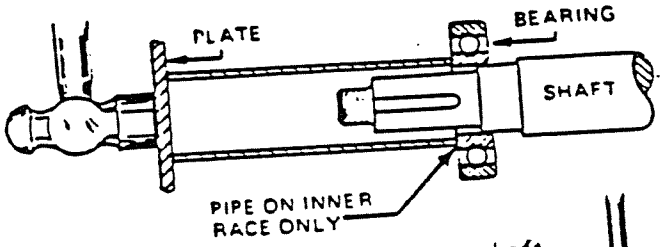
FAN DOES NOT OPERATE

ELECTRICAL OR MECHANICAL

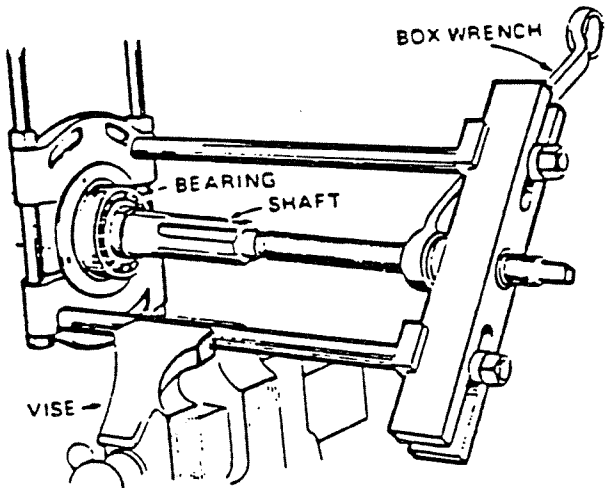
Mechanical and electrical problems are usually straight forward and are analysed in a routine manner by service personnel. In this category are blow fuses, broken belts, looses pulleys, electricity burned off, impeller touching scroll and motor too small and overload protector has broken circuit.

BEARING REPLACEMENT

If a fan shows an increase in vibration, it generally indicates that the bearings require replacing or servicing. When bearings are to be replaced, they should be removed from the shaft with a bearing puller and the same tool can, by relocating the jack nut, be used on site to install the bearing. Always check the fan shaft for wear before replacing the bearing and if the shaft is worn, it too must be replaced. After replacement of bearings, run the fan without a load for a short period to allow new components to bed in.



Driving a bearing onto a shaft.



Using a bearing puller.

END OF SECTION 2

Additional and more detailed information pertaining to *SECTION 2* will be found in the following:

AIR CONDITIONING ENGINEERING 2ND EDITION S.I. UNITS
Whitstable Litho Ltd. Whitstable Kent.

T.P.C. TRAINING SYSTEMS (Air Handling Systems)
Technical Publishing Company, Barrington. Illinois.

ASHRAE SYSTEMS AND EQUIPMENT 1967
American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., New York.

START, TEST AND BALANCE
National Joint Steamfitters - Pipefitters.

HOW TO DESIGN HEATING, COOLING COMFORT SYSTEMS - 3 Edition
Business News Publishing Company, Birmingham, Michigan.

VARIOUS MANUFACTURERS INSTALLATIONS, START-UP, SERVICE MANUALS AND BULLETINS
and many other publications.

VENTILATION NR13

Learning Outcome 3
Student Worksheet 3:1

Question 1.

The maximum pressure a fan can develop occurs when the ductwork connected to the fan is:

a) open to atmosphere, or
b) closed off and all openings sealed.

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Question 2.

In an air conditioning installation, where would you most expect to come across a radial blade centrifugal fan?

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Question 3.

Describe and show, by means of a sketch, the difference between a *tube axial* and *vane axial* fan.

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Question 5.

Give the formula and state the definition which explains what happens to the pressure in a constant duct system if the air flow is varied.

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Question 6.

If, when starting a fan on a new system, the flow rate was found to be high and the static pressure lower than calculated, would this be common or an uncommon occurrence? Give reasons for your answer.

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Question 7.

Explain fully what is meant by the *term system characteristic curve* as used in fan applications.

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Question 8.

A fan is required to supply 8.5 m^3/s when operating at 650 RPM against a static pressure of 850 Pa. On start-up, the capacity is found to be 9.75 m^3/s and the static pressure is measured to be 565 Pa. A 4.5 kW motor is installed and the absorbed power of the fan is 4.7 kW.

i. How would you fix the problem?

ii. What would be the static pressure for the fan at the correct flow rate?

iii. Does the motor need to be changed once the flow rate is corrected?

iv. What is the absorbed power for the fan at the correct flow rate?

SHOW ALL FORMULA AND CALCULATIONS

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CALCULATIONS

Question 9.

What is meant by the term "surge" as applied in fan terminology and how can it be corrected?

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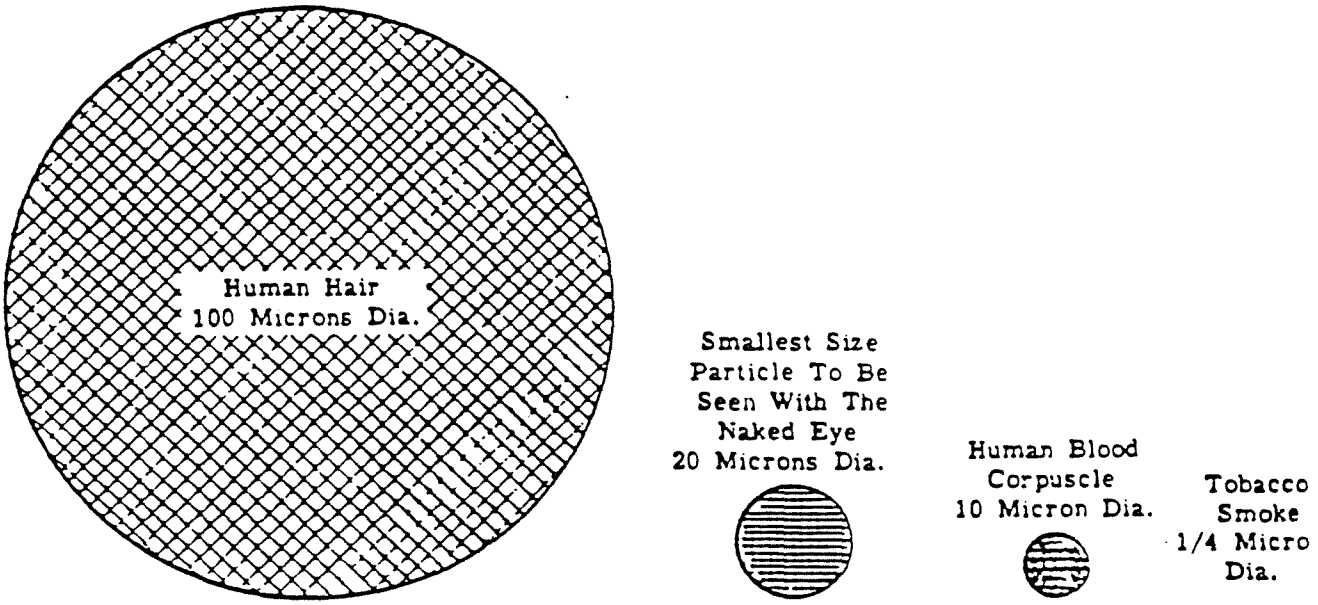
AIR CLEANING ASSEMBLIES

1. One of the concepts of air conditioning is that the air introduced into the zone should be clean. Some of the advantages of air being cleaned is that dirt removed by the filter keeps the inside of the building cleaner and clean air passing over a dehumidifying coil prevents the moisture and dust forming a muddy coating on the coil reducing its heat exchange efficiency and increasing its pressure drop. Increased pressure drop will result in less air flow, again reducing the heat exchange efficiency, and causes more power to be absorbed by the fan's motor/s.

2. Airborne particles are measured in *microns*, a unit of measurement which is defined as one millionth of a metre. For comparison, consider the following:



Greatly Magnified View Showing Relative Particle Size

3. Operating characteristics and the performance of an air filter are described or stated by a number of parameters.

   (a) *Rated Capacity:* is expressed in cubic metres per second (m^3/s) is the recommended maximum rate at which air should be passed through a filter in service. Rated face velocity expressed in metres per second (m/s) is an alternative method of deriving capacity. Capacities are sometimes expressed in litres per second (l/s) 1000 l/s = 1 m^3/s.

(b) *Resistance*: expressed in Pascals (Pa) is the pressure drop developed across the filter as air passes through it. In most types of filters, the resistance is also a function of the amount of dirt in the filter. As the filter *loads up* with dust, the resistance increases. It is normal to quote the *clean resistance* for fixed panel type filters, but for automatic self cleaning filters, the *average operating resistance* is generally used.

(c) *Dust Holding Capacity*: is the mass of dirt in the filter which will increase the resistance at rated capacity either to or through a pre-determined valve of resistance and this is a more significant figure than clean resistance. The dust holding capacity is normally given in *grams*. Velocity also affects dust holding capacity.

(d) *Efficiency*: or more correctly arrestance efficiency is generally defined as the amount of matter retained by the filter expressed as a percentage of the amount entering it. Both atmospheric air and test dusts are used in filter testing standards. As to the measurement of the *amount* of dirt retained by the filter there are a number of ways in which this can be assessed.

(i) Gravimetric or mass test - the amount of dust is determined by weighing the dust and the filter collecting it.

(ii) Dust-spot, blackness or discolouration test, all meaning the amount of dust being measured by assessing the soiling of filter papers through which the air has been passed.

(iii) Count tests in which the number of particles is counted by sampling microscopic examination or by automatic particle counters.

(iv) Opacity test in which the opacity of an air stream containing dust in suspension is assessed.

(e) *Filter Life*: or the time a filter will operate without attention is a function of four parameters.

(i) Dust holding capacity.

(ii) Gravimetric efficiency.

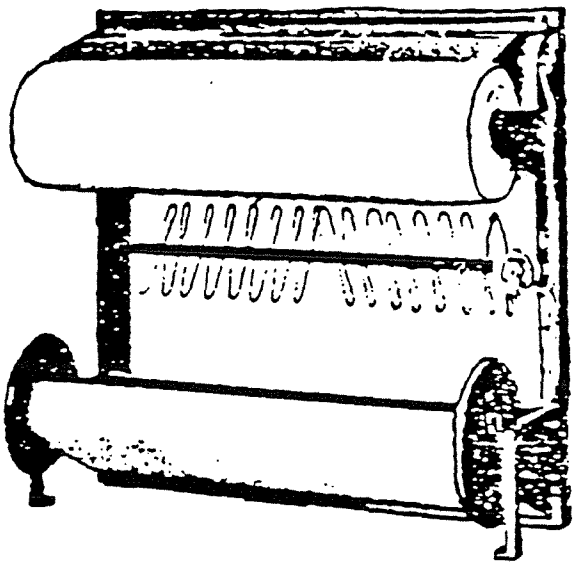
(iii) Dust concentration in air being filtered.

(iv) Rate of air through filter.

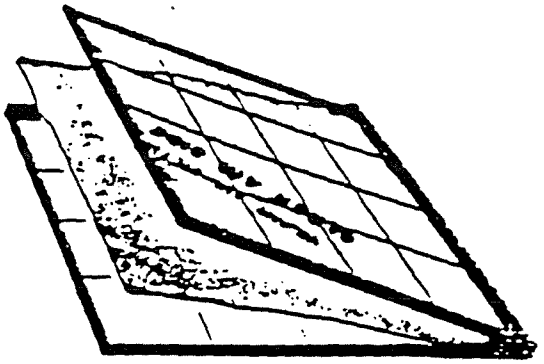
The mass of dust being fed to the filter is the product of the dust concentration and the air flow rate. The mass retained by the filter is the mass fed multiplied by the gravimetric efficiency. *The filter life can be calculated from this figure and the filters dust holding capacity.*
Often when long periods of time between servicing, or when dirt concentration is high, filter life can be increased by decreasing the rate of air flow through the filter.

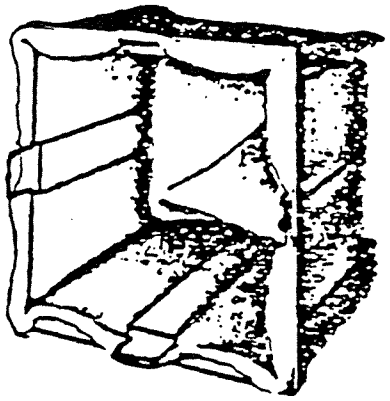
4. Although there are many different types of filters available, they all fit into one or a combination of three basic categories:

(a) *Dry Arrestance*: filters remove dust particles from the air stream by trapping them between the fibres of a filter mat, which may be of cellulose, cloth, felt, glass, paper or specially developed synthetic materials. Dry arrestance filters are available in both the fixed panel (shown below) or roll types (shown opposite).



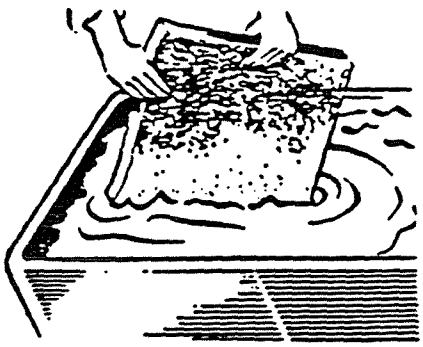
roll type dry media filter





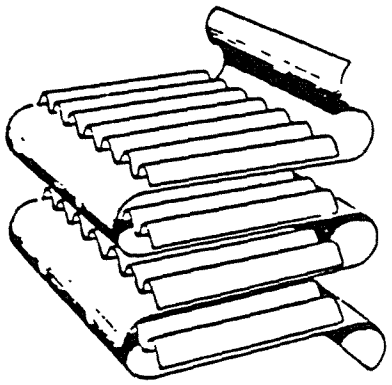
Pyracube deep bed filter

The media texture necessary for high efficiency requires low surface velocity to keep the resistance low. To obtain this low velocity, the area of the filter can be increased within the filter frame either by pleating or forming it in corrugations or by shaping and sewing it into a semi-supporting basket.
This increases the depth of the filter and also increases the effective surface area up to four times the filter face area depending on the configuration.



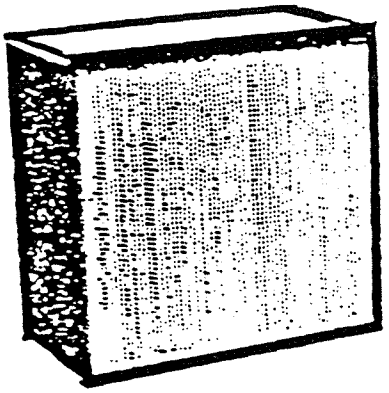
Immerse and agitate in bath of warm or cold water with detergent

All the dry media filter mentioned so far can be cleaned by vacuum cleaning, washing or some other method. However, a dry media filter which is never cleaned is the *Absolute* or *HEPA* (High Efficiency Particulate Air) filter.



Part of an absolute HEPA filter pack, showing corrugated separators

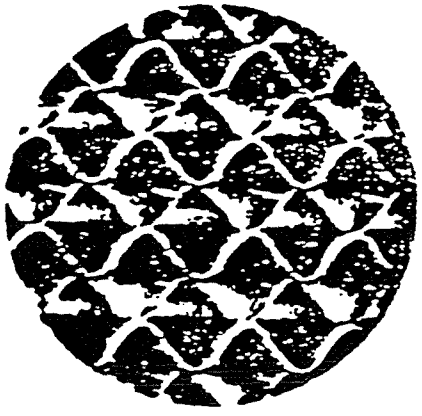
These use a glass paper media and have an efficiency of 99.97%. The paper is arranged in an extreme *concerting* arrangement separated by pieces of corrugated material, generally aluminium. The filter pack of paper and separators are sealed in a strong frame generally made of metal.



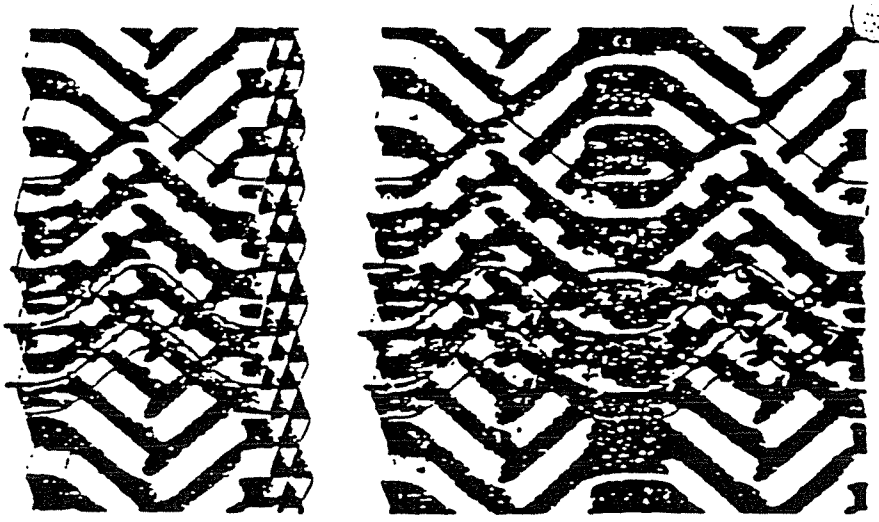
high efficiency filter

(b) *Viscous:* All the dry media filters, with the exception of the *HEPA* filters, can be impregnated with an adhesive, normally a gel or light oil, to increase their efficiency. Generally medias impregnated with an adhesive cannot be satisfactorily cleaned.

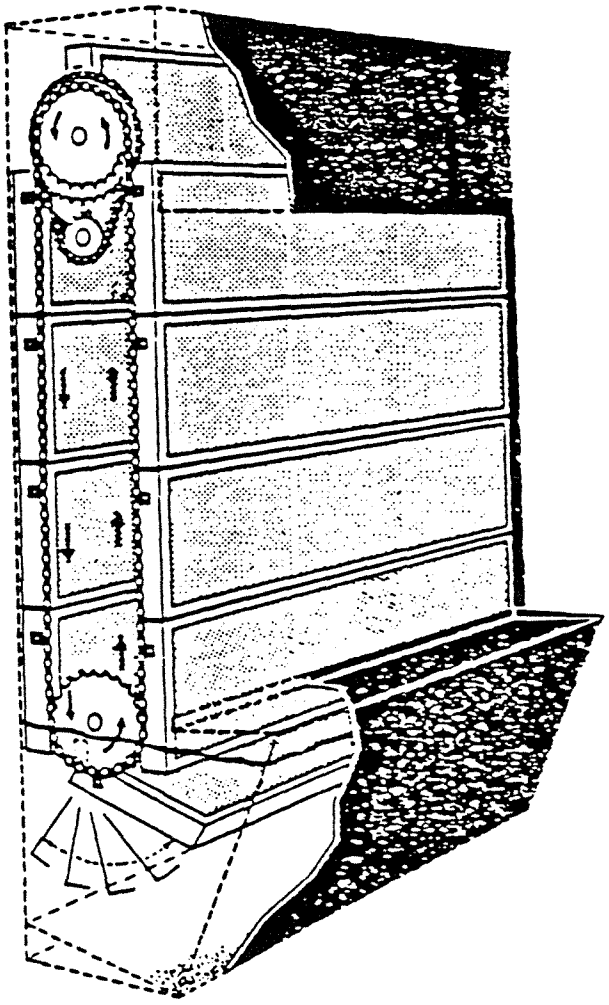
Metal viscous oil filters are available as simple fixed panel filters or as automatic types which provide for the panels to be cleaned and re-oiled by passing them through an oil bath in the base of the filter frame. The filters are normally constructed from aluminium.



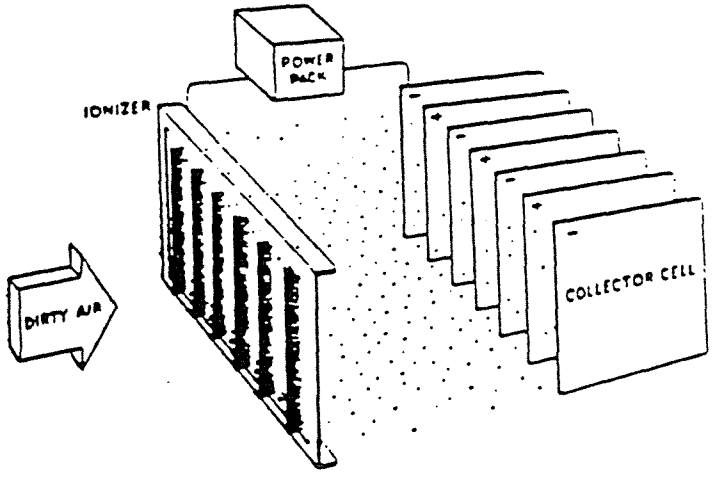
Section of filter media (approx. actual size)



Cross section of filter media showing airflow pattern



(c) *Electrostatic*: filters utilize the principle that unlike electrostatically charged bodies are attracted to each other. The air to be cleaned is first passed through a unit called the *ioniser*, which consists of a series of earthed electrodes having fine tungsten wires spaced centrally between them. When a high voltage D.C. supply is connected between the earthed electrodes and the wires, a very high voltage gradient is produced around the wire and a vast number of ions are generated in the intervening space.

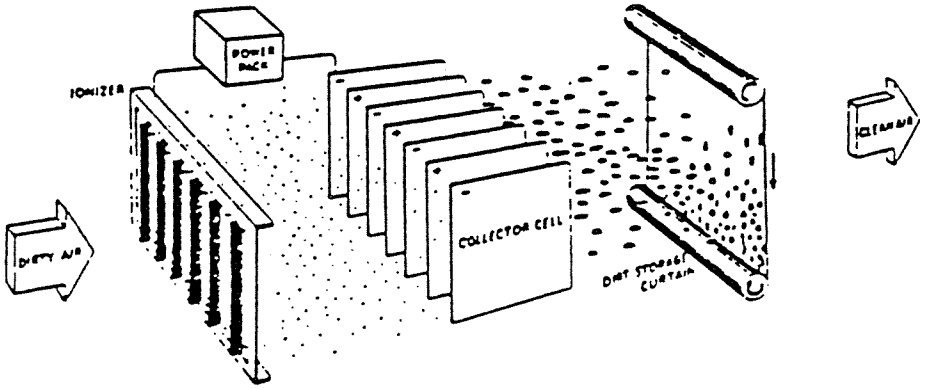


Principle of operation of Ionitron electronic air filter

As the airborne dust particles pass through this highly ionised space, they are given a specific electrostatic charge. The charged particles then pass through a collector cell where an electrostatic field between charged plates forces them towards a plate of opposite polarity where they adhere. An adhesive is applied to the plates to ensure retention. The unit is cleaned by flushing away the adhesive and precipitated dirt with water. Fresh adhesive is then applied before placing the filter back in service.

Direct Current at high voltage, 13 kV for the wires and earthed plates and 6.5 kV for the collector cells parallel plates, which are alternatively earthed, is supplied from a 240 volt, 50 cycle, single phase power pack supply.

A variation of the electronic filter is the *Ionitron Agglomerator*. An agglomerator combines an ioniser and cell fed with high voltage direct current from a power pack but the cells are not coated with adhesive. In addition, there is a dirt storage section immediately following the collector cells.



Principle of operation of Ionitron agglomerator type electronic air filter.

The ionised dust is precipitated onto the cell plates and here it *agglomerates* into large particles and as there is no adhesive, gets blown off again. The large agglomerated particles are then carried into the dirt storage section where, because of their large size, they are effectively 100% trapped.

NOTE: *Any maintenance or repair of electronic air filters must be done with the power OFF. Do not attempt to bypass any of the electrical safeguards installed.*

5. Since the effectiveness of an air conditioning system is dependent on the amount of air flowing, care should be taken to ensure that the resistance of the system does not materially increase. Draught gauges installed across the filter will provide a continuous reading of the filter resistance. The point at which service is required alters depending on system or manufacturers recommendations. As a general rule, filters should be serviced before the total air flow in the system drops by 10% and/or the pressure drop across the filter rises to 250 Pa.

6. In some locations the odour of air may need to be controlled as well as the cleanliness. *Activated Carbon*, which is a special form of carbon, is normally used for odour control. Activated carbon is capable of *soaking up* or adsorbing gases and vapours, just as silica gel adsorbs moisture. The air is passed through beds of activated carbon, when the carbon is saturated, it can be rejuvenated by heating to 540^{0}C.

7. There is nothing to prevent the installation of different types of filters in series in the one air duct. An example is the use of a panel type dry fabric filter as a pre-filter to remove the larger particle of dust leaving a HEPA filter to collect only the finer material. A second example is the installation of an after filter in an *electrostatic precipitator*, again normally a dry fabric panel type, in case of power failure.

VENTILATION NR 13

LEARNING OUTCOME 3

STUDENT WORKSHEET 3:2

Question 1.
List the three categories of filters.

(A) _____

(B) _____

(C) _____

Question 2.
Which filter is the most efficient and what type of media does it use?

_____

_____

_____

_____

Question 3.
How many types of electric filters are there ?

_____

_____

_____

Question 4.
How would you check if a filter required cleaning or replacing?

_____

_____

_____

_____

Question 5.
What metal is used in the manufacture of Metal Oil Filters?

_____

_____

Question 6.
What precaution must be observed before working on any electronic type filter?

Question 7.
How is odour generally filtered out of the air in an air conditioning system?

Question 8.
What do you expect to happen to a Direct Expansion coil if the filters are:

(A) Removed from the system for a reasonable period of time?

(B) Allowed to clogg up?

LEARNING OUTCOME 4

Assessment: Short answer, practical exercise.

Performance:

a. Briefly describe the correct operating procedures for a range of test equipment, commonly used on ventilation equipment, and state how the information gathered could be used.

b. Use the most appropriate items of test equipment to take the following measurements:
- air Volume
- air Velocity
- air Pressure
- sound power level

from various points on a ventilation/air conditioning system.

c. Whilst carrying out the above procedures the student will explain to his/her lecturer how they have arrived at their measurements, what they consider to be the most appropriate units to use and how the above measurements are related.

TEST EQUIPMENT

- Practicality
- Types available
- Usage and care
- Calibration
- Flow characteristics
- Appropriate formulas
- Units of measurements
- Application
- Use of information gathered
- Permanently installed/portable
- Expected results

Suggested teaching time: 4 hours.

NOISE/VIBRATION

- Source of noise
- Nature of sound
- Frequency
- Acceptable levels
- Measurement
- Transmission
- Controlling noise/vibration
- Attenuation
- Absorption
- Fault finding

Suggested teaching time: 4 hours.

AIR BALANCING INSTRUMENTS

Before you can attempt to balance an air stream, you must have accurate, reliable instruments. *All instruments with the exception of the pitot tube and manometer, need to be recalibrated at intervals of not more than six months.* If an instrument is dropped, or otherwise damaged, it should be recalibrated. If there is any doubt about the accuracy of an instrument, it should be checked against another instrument measuring the same quantity and be recalibrated if the answers do not agree. *Each instruments should carry a calibration record.*

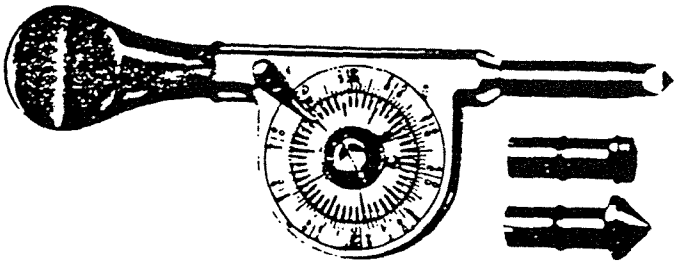
Read the manufacturer's instructions and become familiar with your air testing instruments before using them on an actual job.

1. *ROTATIONAL SPEED MEASURING INSTRUMENTS*

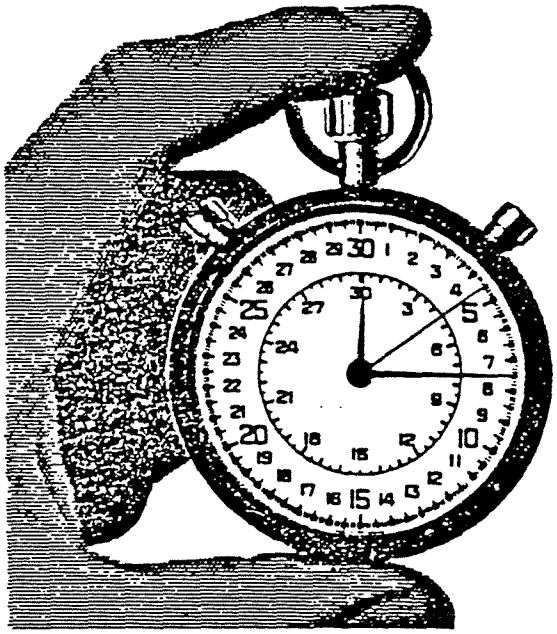
Measurement of rotational speed will be required to determine whether a fan is working as specified. Mechanical *tachometers* are normally used where the shaft of the fan or motor is accessible. Where it is dangerous or not possible to use a mechanical tachometer, a *stroboscope* is usually employed.

(a) Revolution Counter (Speed Indicator)

A small, inexpensive, hand held counting device that is pressed to the centre of a rotating shaft for a timed period. It requires the use of a stop watch and cannot usually be reset to zero.



(A) Speed indicator with several tips



(B) A stop-watch should be used for timing when reading RPM with speed indicator

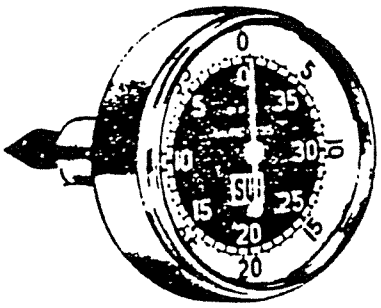
Speed indicator

$$R.P.M. = \frac{Nf - No}{t}$$

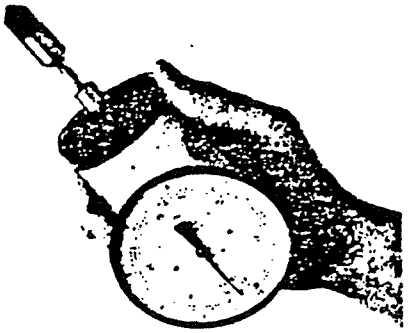
where: $R.P.M.$ = *Shaft speed, revolutions per minute.*
No = *Original reading.*
Nf = *Final reading.*
t = *Time interval, minutes.*

(b) Dial Tachometer (Centrifugal Type)

A hand held instrument that directly indicates the instantaneous speed on the face of a dial.

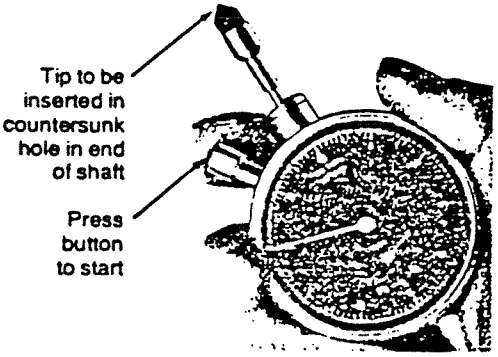


Tachometer with single speed range



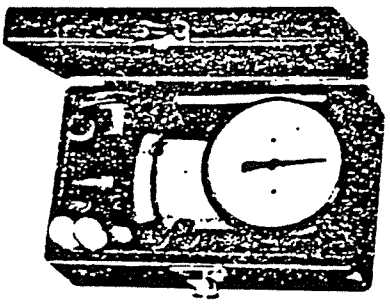
Tachometer with multiple speed ranges

(c) Chromometric Tachometer

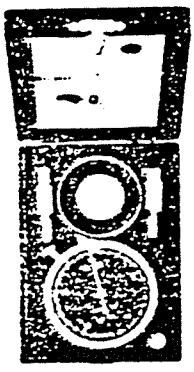


A hand held instrument that combines a precision stop watch and a revolution counter in an instrument. The scale is calibrated so that after 6 seconds the instrument stops accumulating revolutions. Each actual revolution of the instrument indicates 10 revolutions on the dial, so that readings are directly in R.P.M. *Repeatedly pressing the activating button without contact with a rotating shaft, on allowing the timer to run down, will damage this instrument.*

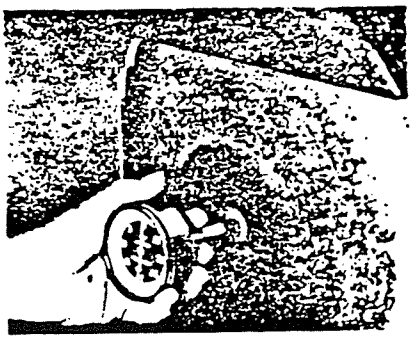
Chronometric tachometer



Always keep tachometer and all attachments in case when not in use



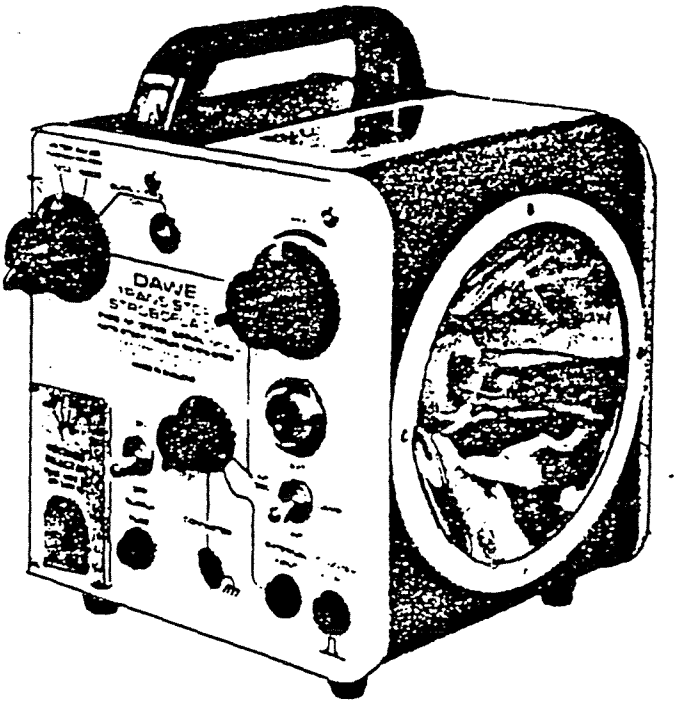
Complete tachometer set with extra tips and case



Reading shaft speed

(d) Stroboscope

This instrument is a source of variable frequency light and the technique in using it is to match the light frequency to the rotational speed. A chalk mark is made on the pulley and the stroboscope adjusted until the mark appears to be stationary when the pulley is rotating. The frequency setting on the stroboscope may, however, be a multiple or sub-multiple of the actual shaft speed and such errors of harmonics need to be eliminated to assess the correct speed. *The stroboscope is normally used on site only when it is impossible to use a mechanical tachometer.*



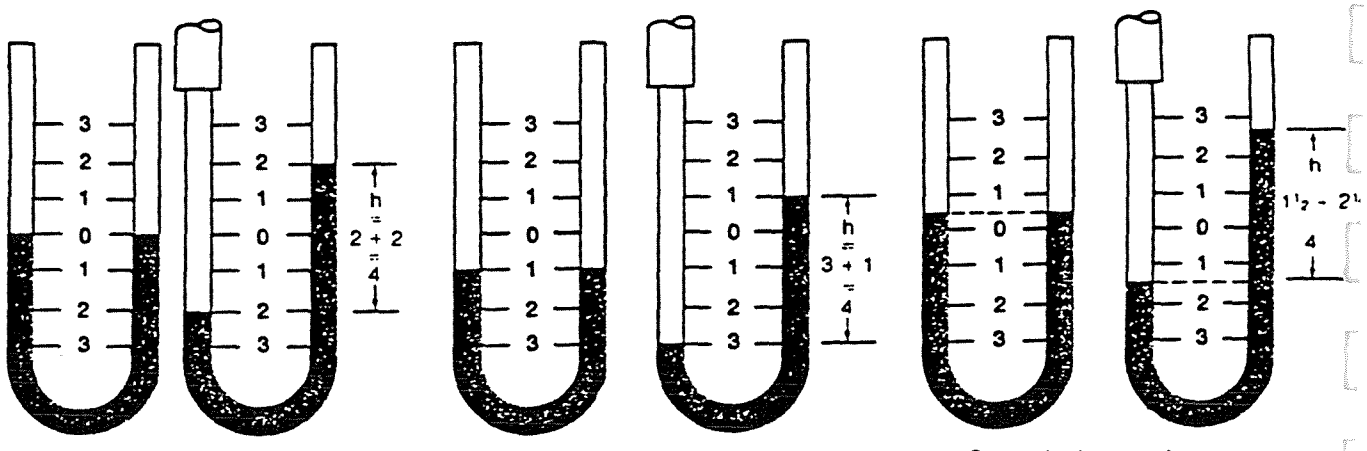
Stroboscope

2. PRESSURE MEASURING INSTRUMENTS

For in duct measurements a pitot tube with an adjustable manometer will give the most reliable results. These need no calibration and are consistent. In practice the lower pressure readability on the manometer is about 10 Pa (4 m/s or 1.0 mm).

(a) U-Tube Manometer

This type of manometer is a simple and useful instrument for the measuring of partial vacuums and pressures, both for air and hydronic systems. In its simplest form, the manometer consists of a U shaped glass or plastic tube partially filled with liquid.



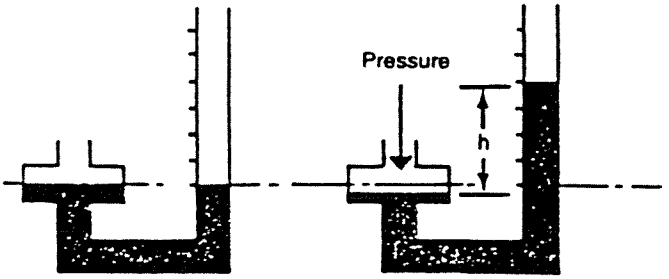
(A) Levels at zero of scale (B) Levels below scale zero (C) Levels above scale zero

Reading the U-tube manometer

A difference in height of the two fluid columns denotes a difference in pressure in the two legs. When reading a U-tube manometer, observe that the liquid level in one leg falls while the liquid in the other leg rises. The pressure h is the *sum* of the readings at these two levels.

(b) Well-Type Manometer

As described previously, the U-tube manometer requires two readings. The well type manometer requires only that one reading be taken. The scale is corrected to compensate for the change in reservoir liquid level so that the well does not have to be excessively large.



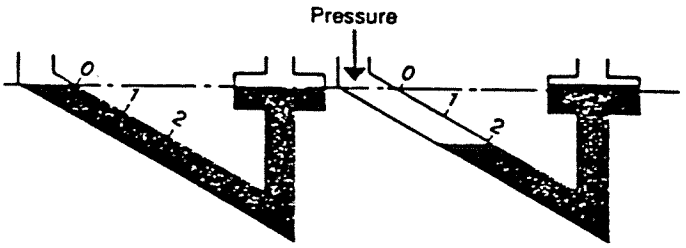
(A) Equal pressure (atmospheric) on both legs (B) Pressure imposed on well

Well-type manometer

(c) Inclined Manometer (Draft Gauge)

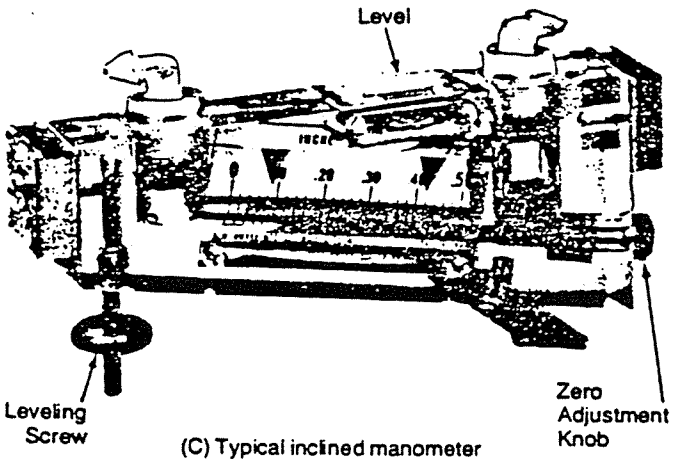
The inclined manometer is often called a draft gauge because of its use in measuring the draft in boilers.

The inclined manometer is a variation of the well type, but the indicator leg is placed in a sloping position rather than vertical. The purpose of this is to improve accuracy in reading the scale. *For the same pressure the distance of the incline scale is considerably greater than the vertical scale.*



(A) Equal pressure (atmospheric) on both legs (B) Pressure imposed on inclined leg

The inclined manometer is commonly used for measuring low pressures of 50mm (500 Pa) or less.

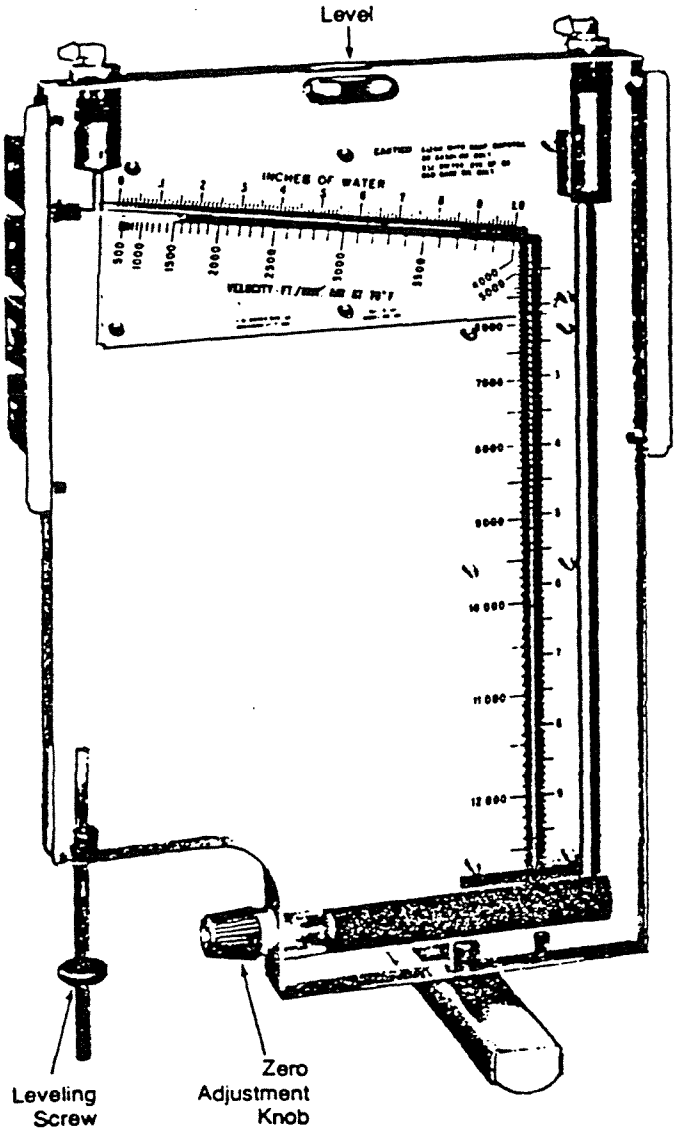


(C) Typical inclined manometer

Inclined manometer

(d) Combination Inclined - Vertical Manometers

Many U and well type manometers may also be used as vertical manometers. This allows for reading both high and low pressures on the same instrument.

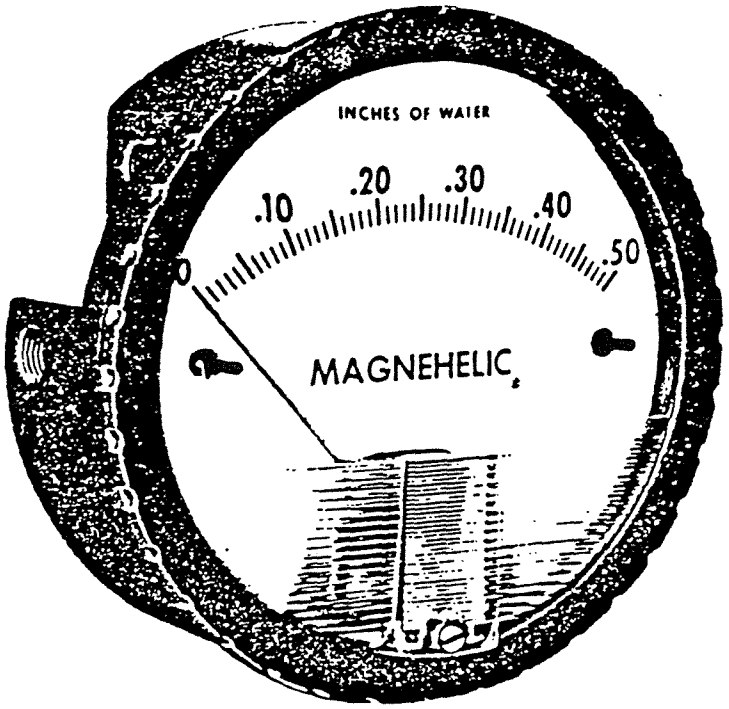


Combination inclined-vertical manometer

NOTE: *In most commercial instruments, the fluid used is a light oil, normally with a specific gravity of 0.83. It is important that only the correct fluid is used in each instrument as the scale is corrected for the designated specific gravity.*

(e) Portable Air Pressure Gauges (Dry-Type)

Manometers have the disadvantage of containing liquid which can be spilled or contaminated, and they must be accurately levelled. Dry type draft gauges overcome these problems. One such operates on a bellows that actuates a helix which, in turn, moves a pointer across a printed scale. Others operate by weight or blade displacement. Generally, dry gauges are not as accurate as a liquid filled U-tube and frequently do not stand up to rough handling. Several instruments are required to cover the normal ranges encountered in average air conditioning jobs. There are approximately 30 available pressure ranges in this instrument.

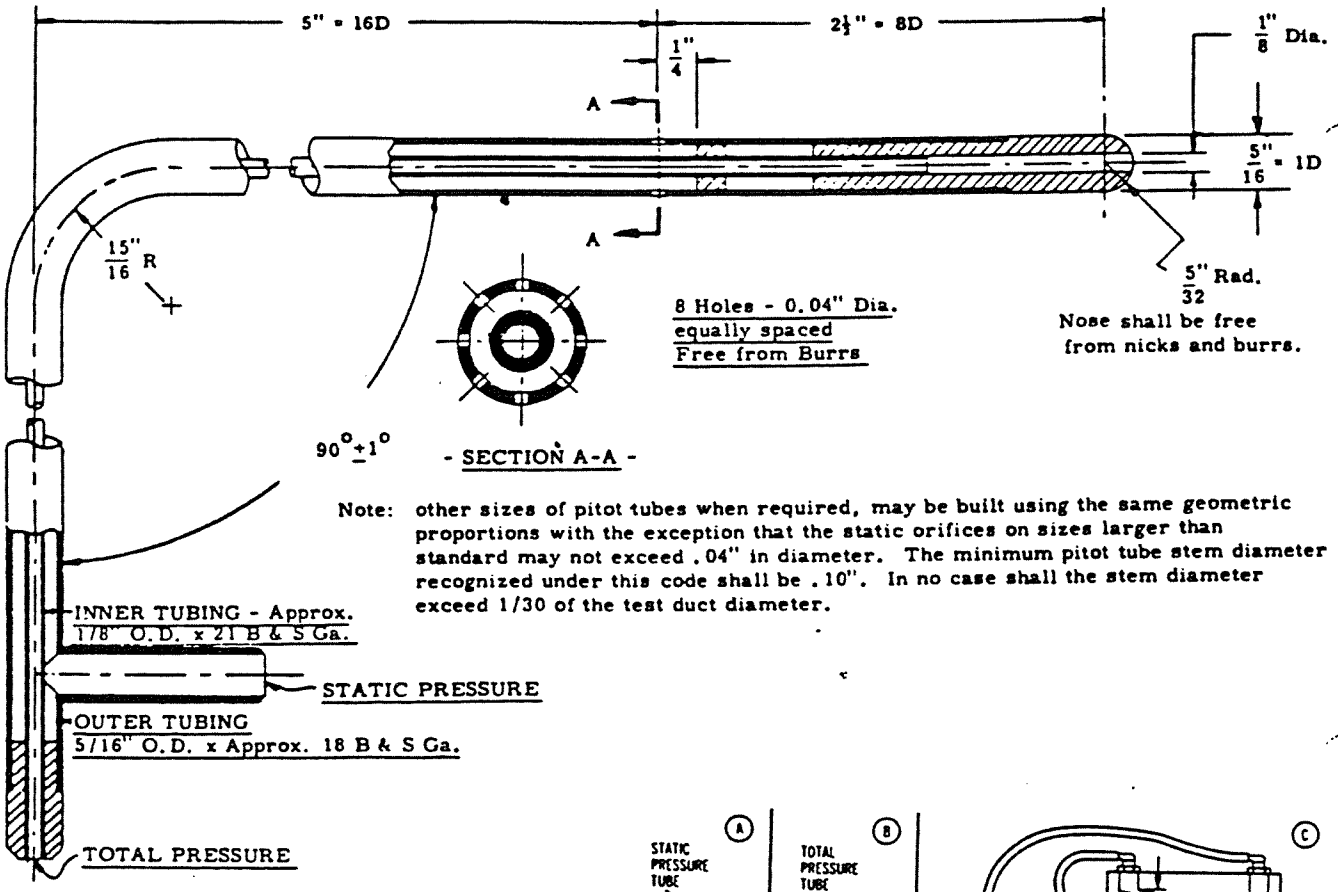


Dry type draft gage

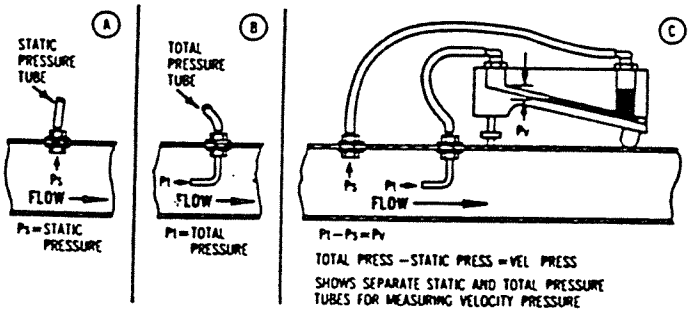
3. *PITOT TUBE*

The Pitot tube (named after the Frenchman, Pitot, who designed it) is an instrument for measuring *Static, Velocity* and *Total* pressure in a stream of air or gas.

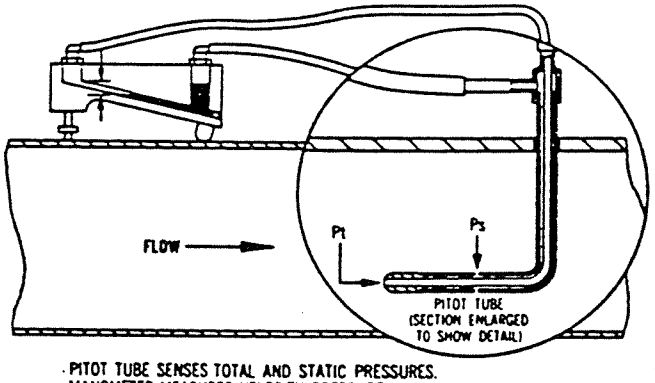
It consists of two concentric tubes, usually *"L"* shaped for convenient handling and which ends in *two* separate outlets for connection to a manometer. The instrument is inserted into the air stream, parallel to the direction of flow and with the openings always pointed upstream. The *inner* tube, facing into the air stream, conveys the *impact* or *total* pressure. The *outer* tube has a number of small radial holes in its wall and the annular space between the two tubes conveys the *static* pressure.



Note: other sizes of pitot tubes when required, may be built using the same geometric proportions with the exception that the static orifices on sizes larger than standard may not exceed .04" in diameter. The minimum pitot tube stem diameter recognized under this code shall be .10". In no case shall the stem diameter exceed 1/30 of the test duct diameter.

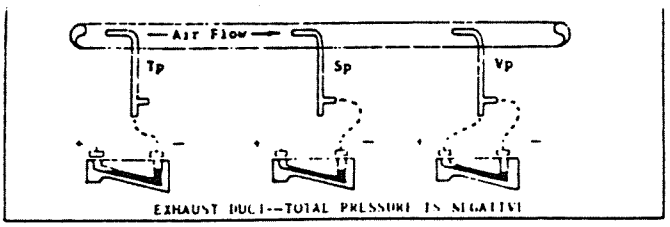


TYPES OF PRESSURE MEASUREMENTS

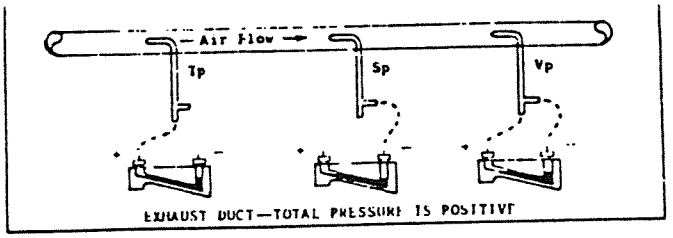


PITOT TUBE SENSES TOTAL AND STATIC PRESSURES.
MANOMETER MEASURES VELOCITY PRESSURE —
(DIFFERENCE BETWEEN TOTAL AND STATIC PRESSURES).

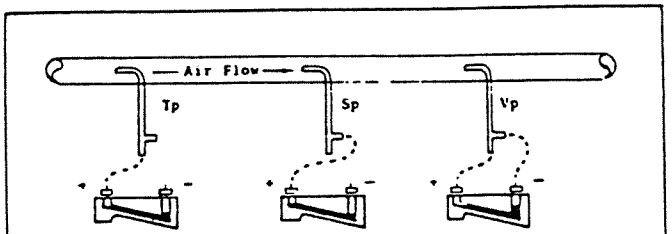
Pitot tube hook-ups are shown in the following figures. It can be seen from these figures that whatever the condition, the hook-up for reading *velocity* pressure remains unchanged; *the impact (total) tube goes to the high pressure side and the static tubes goes to the low pressure side.*



EXHAUST DUCT—TOTAL PRESSURE IS NEGATIVE



EXHAUST DUCT—TOTAL PRESSURE IS POSITIVE



SUPPLY DUCT — TOTAL PRESSURE IS POSITIVE

DETERMINATION OF FLOW VOLUME

The object of an actual test usually is not only to determine the flow velocity at a certain point, but to determine the volume of air flowing through a duct.

EXAMPLE

From a 10 point traverse of a 300mm round duct the velocity pressures are tabulated and corresponding velocities found and also tabulated, as shown in the following table, the average velocity being found as the average of all velocities.

Horizontal		Vertical	
mm	m/s	mm	m/s
1.3	4.6	1.0	4.0
1.8	5.4	1.5	4.9
2.3	6.1	2.0	5.7
2.8	6.8	3.0	7.0
3.3	7.3	3.6	7.7
3.6	7.7	3.3	7.3
3.0	7.0	3.0	7.0
2.6	6.5	2.8	6.8
1.8	5.4	1.8	5.4
1.0	4.0	1.3	4.6
	60.8		60.4

Average 60.6 m/s

It is necessary to convert each velocity pressure reading into velocity, and then to average these velocities.

An *incorrect answer* will be obtained if velocity pressures are averaged and the result converted to velocity.

Having found the average velocity, the volume of air flowing can be calculated as follows:

Duct size: = 300mm round
Duct Area: = 0.0707 m^2 (approx.)
Velocity: = 60.6 m/s

Volume: = Velocity x Area
= 60.6 x 0.0707
= 4.28 m^3/s

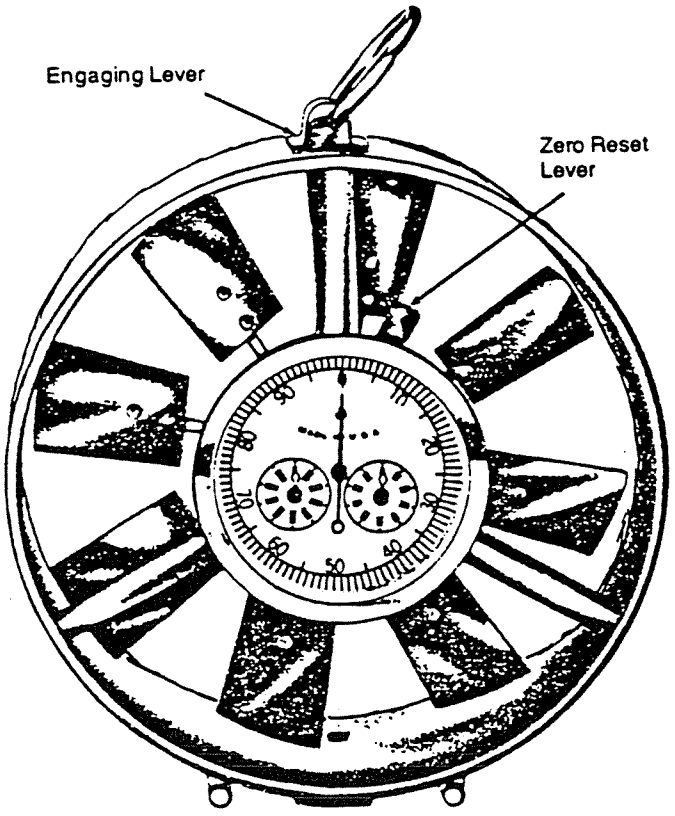
4. ANEMOMETERS

Critical to the proper use of any anemometer is the determination and application of the correction factor which will convert the anemometer reading into a reasonable accurate velocity or volume measurement.

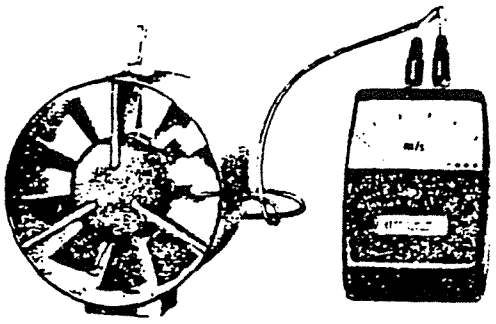
It is absolutely essential that the correction factor be used for each instrument.
The manufacturer tests each type of supply outlet and return inlet to determine the "K-factors" for all of the different types of instruments. The manufacturer also determines the location on the face for making measurements.

(a) Rotating Vane (Propeller) Anemometer

The propeller or rotating vane anemometer consists of a light weight, wind driven wheel connected through a gear train to a set of recording dials that read the linear metres of air passing through the wheel in a measured time. The instrument is made in various sizes; 75mm, 100mm and 150mm diameter sizes being the most common. Each instrument requires individual calibration. Most of these instruments are not sensitive for use below 1.0 m/s. Their useful range being from 1.0 m/s to 10 m/s.
The instrument is available as a manual timed type in which a watch or stop watch is required for timing or with a built-in automatic stop watch.



Rotating vane anemometer



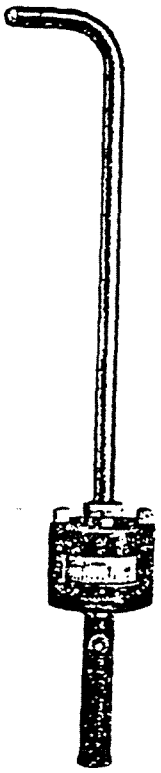
Vane Anemometer with electrical remote transmission

Where remote readings or pulsating flows may be encountered, an instrument with electrical remote transmission is generally employed.

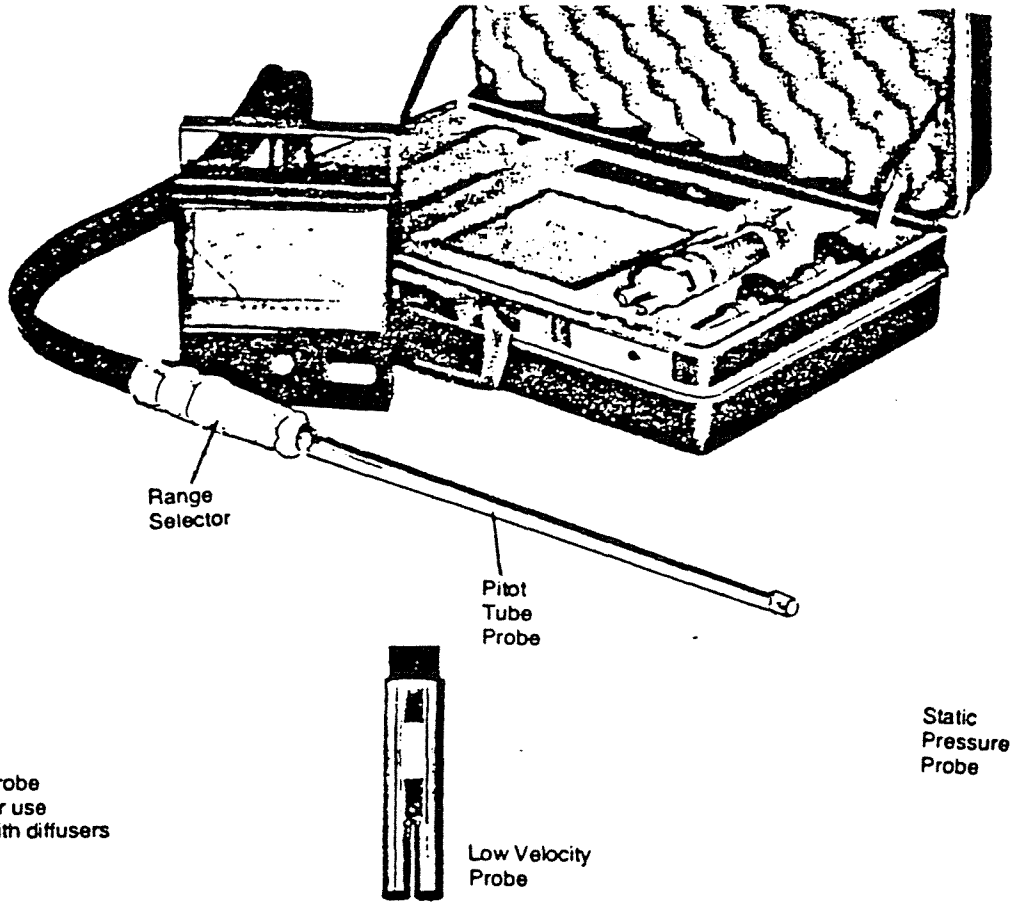
The rotating vane anemometer is used both with fixed stationary readings and travelling averaging time readings. Both give the same accuracy.

(b) Deflecting Vane Anemometer

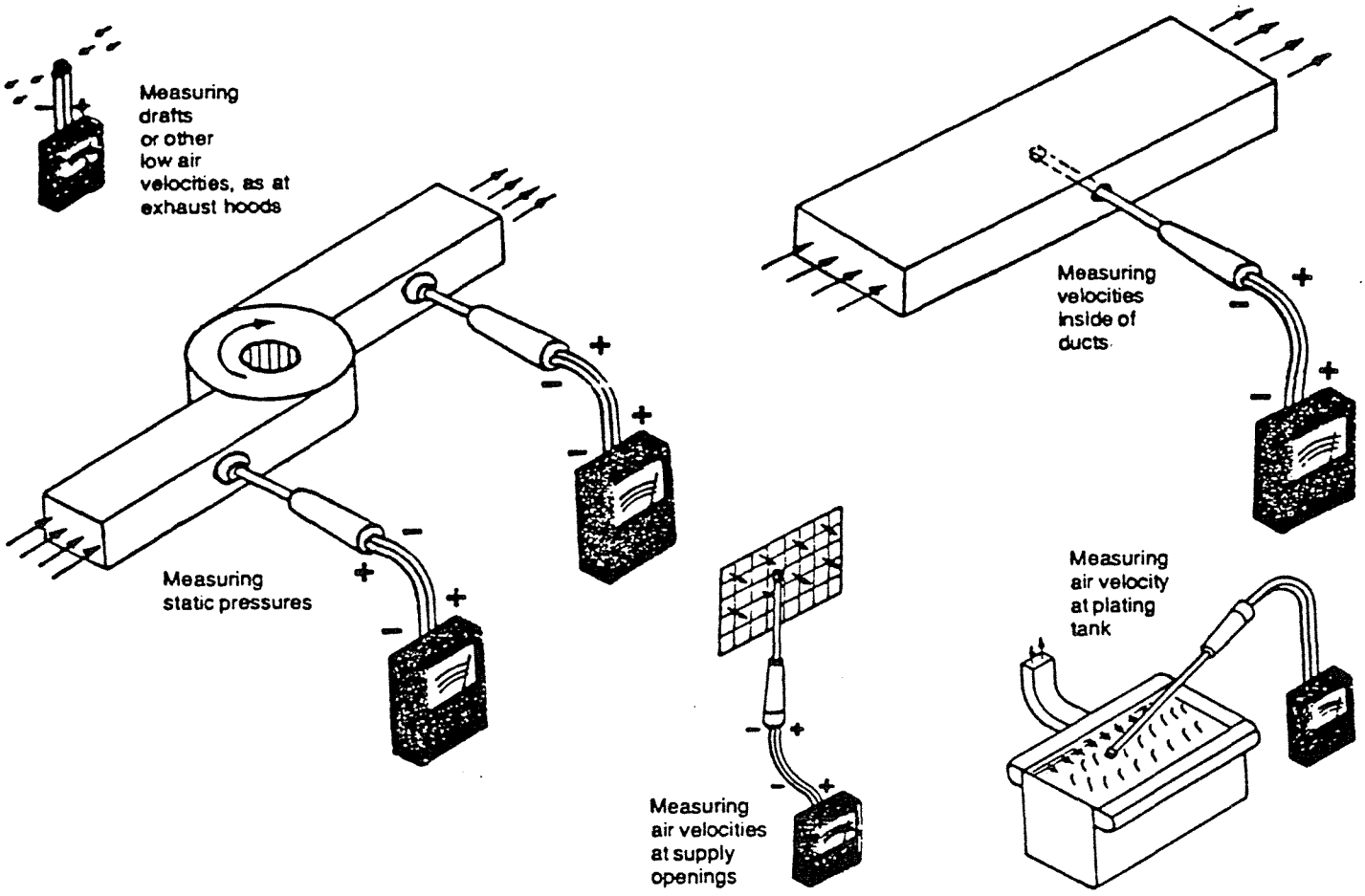
This instrument is commonly known and referred to as a *Velometer.* The deflecting vane anemometer consists of a pivoted vane enclosed in a case. Air exerts a pressure on the vane as it passes through the instrument from an upstream to downstream opening. The movement of the vane is resisted by a hair-spring and dampening magnet. The instrument gives instantaneous readings of directional velocities on an indicating scale. The instrument can be used with various types of remote connected measuring tips or jets - *each of which must be individually calibrated with its individual instrument and tubing.*



Air Flowmeter



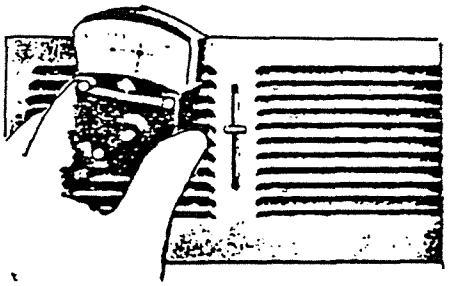
Deflecting vane anemometer



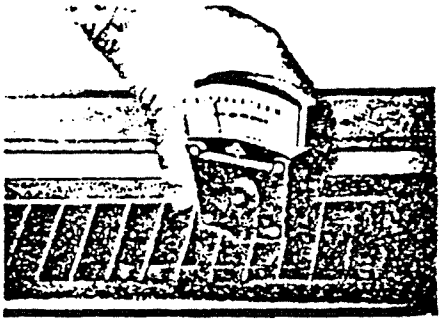
Typical applications of deflecting vane anemometer

(c) *Direct Reading Anemometer (Florite and Bridled)*

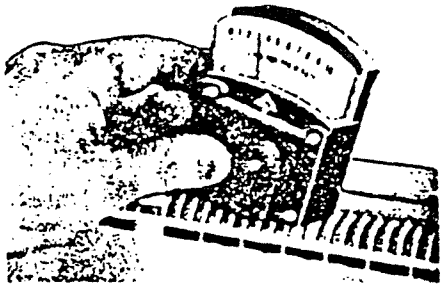
Similar to the already described vane anemometer, however, these include a calibrated scale built into the instrument. The *Florite* type, shown below, is limited in use because of the scale limitations.



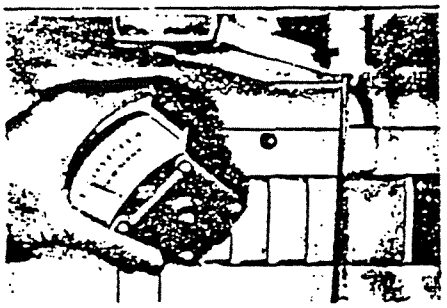
Side Wall Register



Floor Register



Baseboard Diffuser

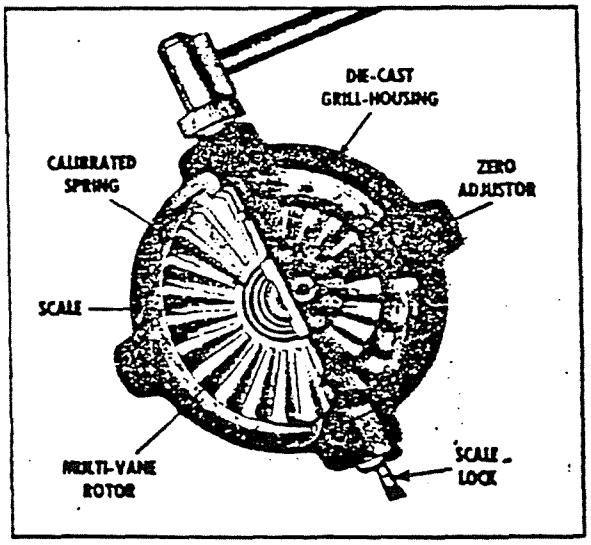


Refrigeration Grille

Pocket type deflecting vane anemometer, and typical applications

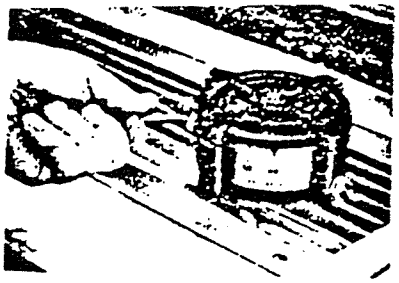
The *Bridled* vane anemometer, shown below, is often used with special attachments that direct air through the instrument. The *Bridled* anemometer is also often used with an extension handle to reach high side or ceiling outlets and is provided with a scale lock which locks the indicator in place once the reading is made.

(A) View showing scale

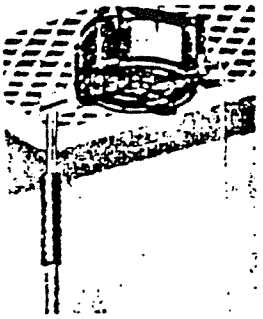


(B) View showing vane

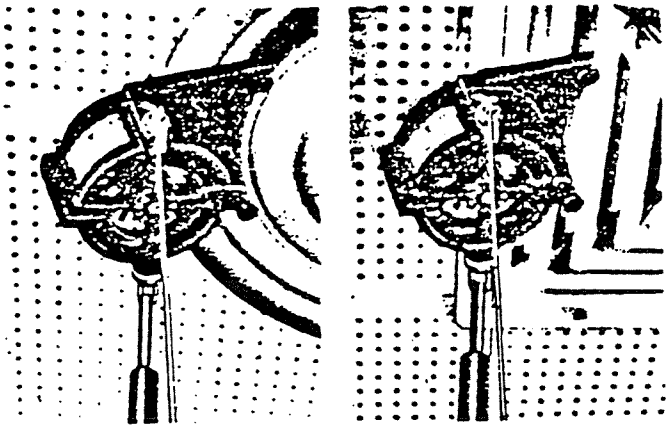
Bridled vane anemometer



Measuring grille velocity: use of handle minimizes interference with air stream



Used overhead: string operates scale lock



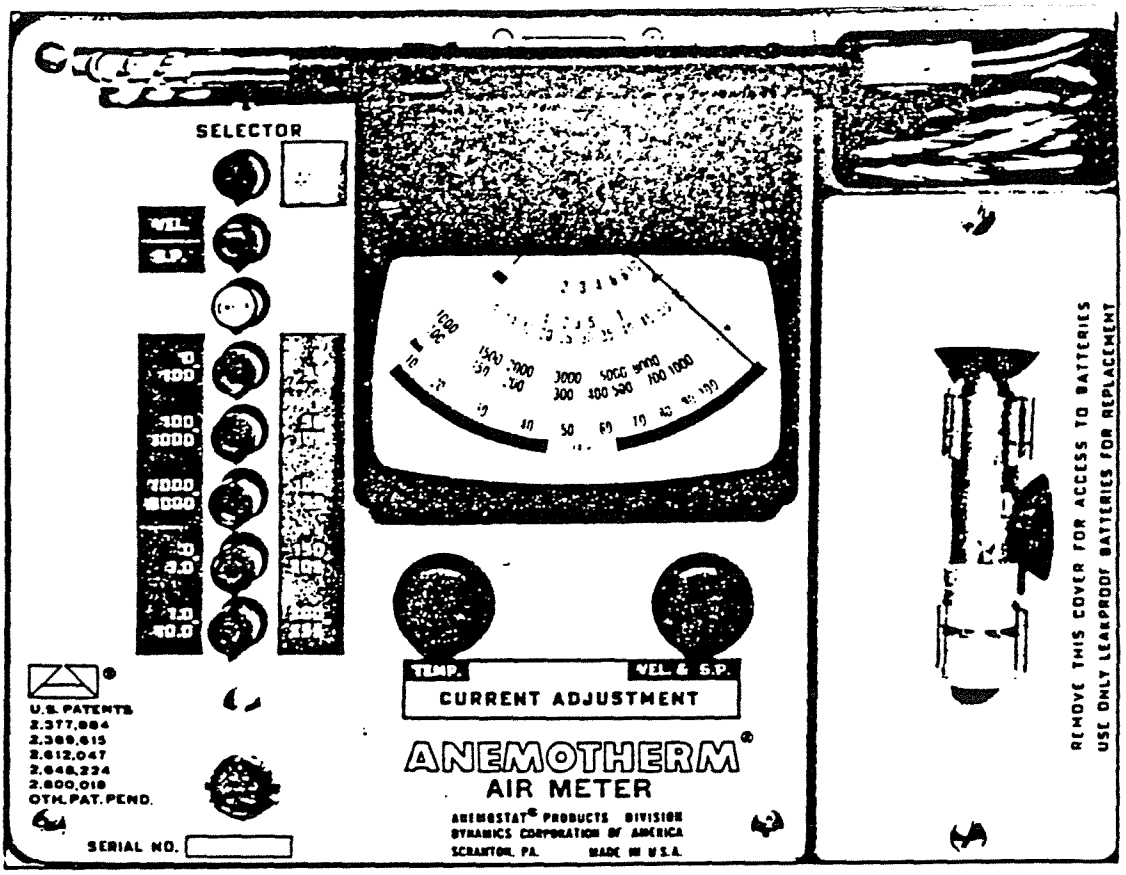
Bridled vane anemometer with air scoop housing used to measure diffuser outlet velocity

Bridled vane anemometer

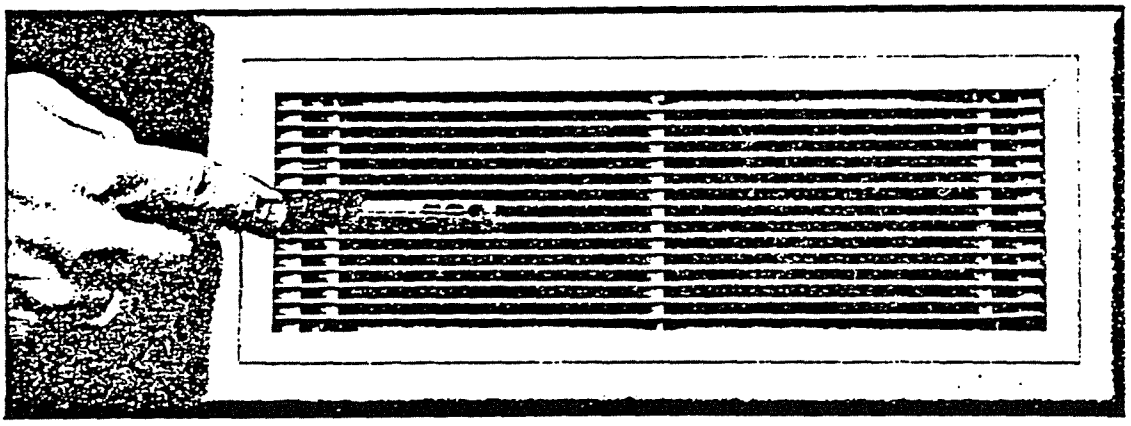
Both the Florite and Bridled vane anemometers require considerable judgement to obtain average velocities, especially when recording at right angles to the air flow.

(d) *Hot Wire Anemometer*

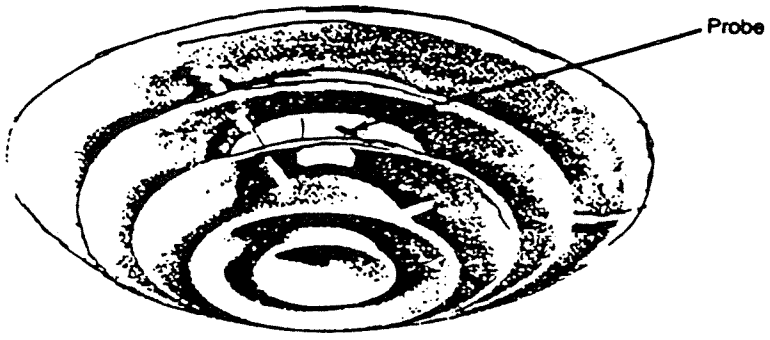
This instrument utilizes the principle that resistance of a heated wire will change with temperature. A probe with a wire element energized by batteries is connected to the instrument. As the air flows over the element in the probe, the temperature of the element is changed from that which exists for still air and this change is indicated as a velocity on the scale of the instrument.



The instrument is also provided with temperature scales that can be read by selecting the correct button. Static pressure can also be measured on some instruments if the proper probe is supplied with the instrument.

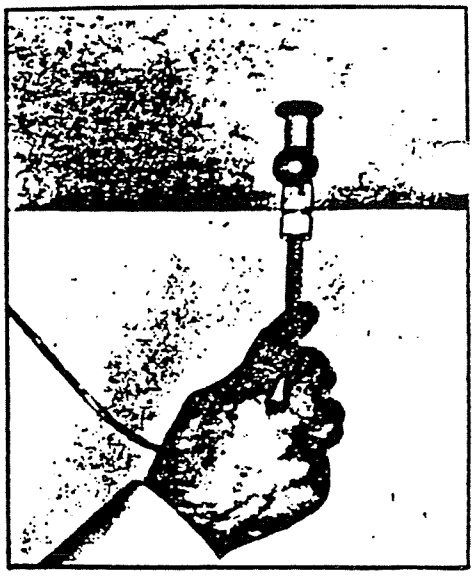


The probe of the "hot wire" anemometer is quite directional and must be used in the precise location recommended by the outlet manufacturer.

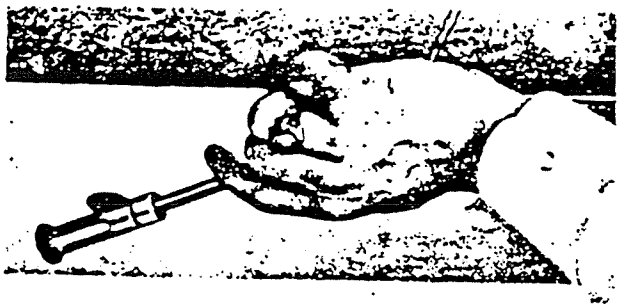


Measuring air velocity in neck of a round diffuser

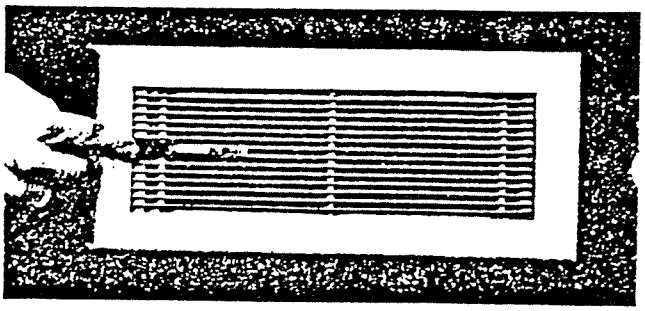
Measuring positive static pressure



Measuring negative static pressure



Measuring grille face velocity



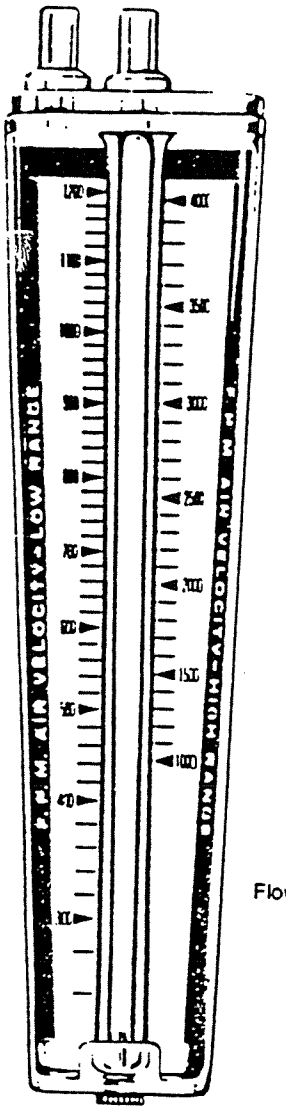
Typical applications of the hot-wire anemometer to air flow measurements

(c) *Variable Area Flowmeter*

The anemometer shown is classified as a *variable area flowmeter*. Air flowing from the probe enters the bottom of the meter where it impinges against the bottom of a small ball. This causes the ball to rise in a tube which is tapered so that the flow area increases at the rises. The height to which the ball rises is proportional to the air velocity, which can be read directly in metres per second on an appropriately graduated scale.

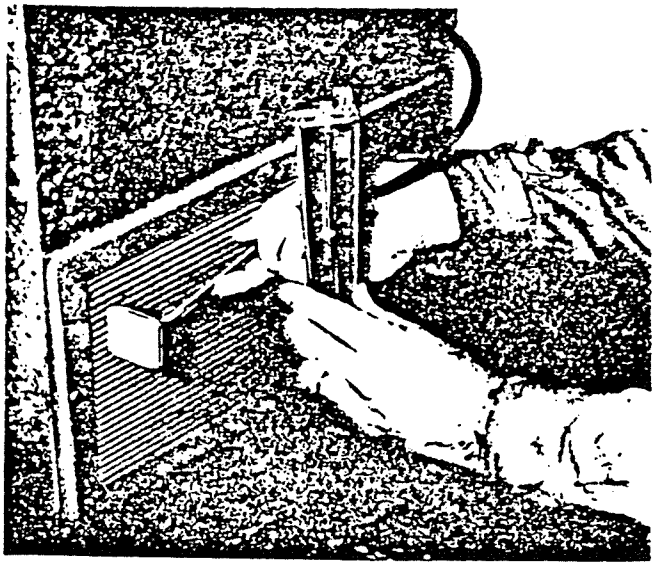
The probe, the instrument tube and connecting tubing comprise a calibrated instrument. It is, therefore, important to use the proper probe, and that the length of the connecting tube not be changed.

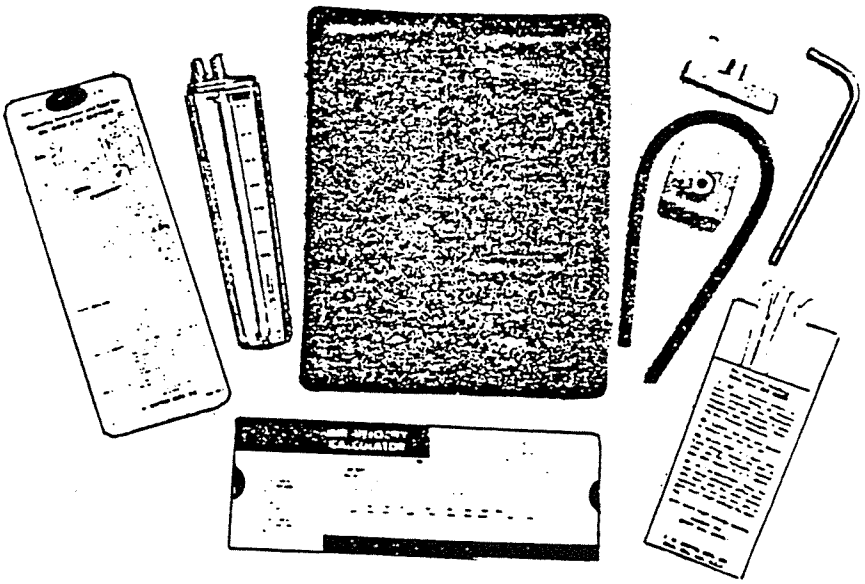
When using a suitable probe, the instrument can be used to measure static pressure.



Flowmeter

Measuring grille velocity



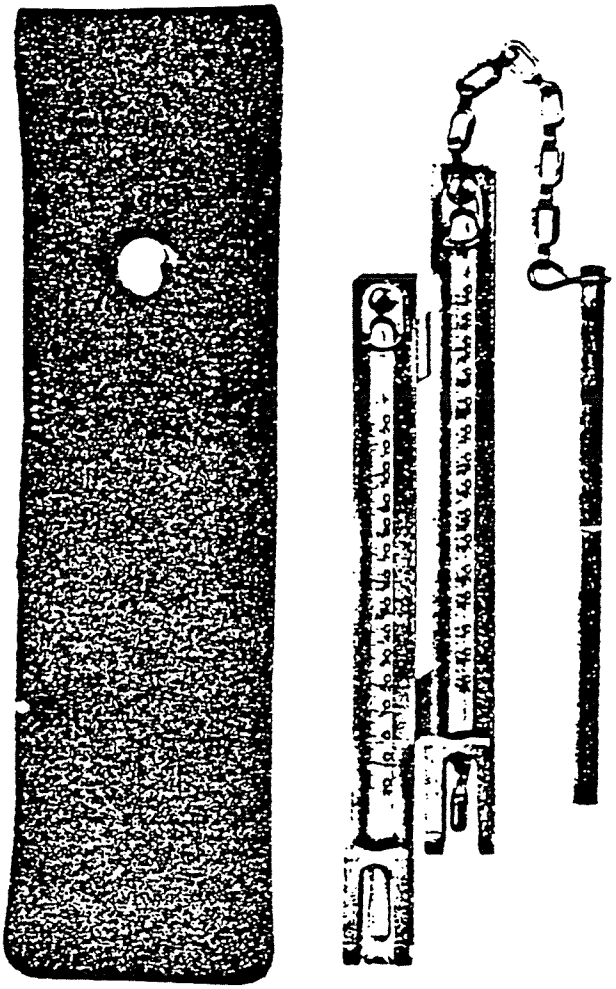


Flowmeter kit with probes, accessories and case

Variable area flowmeter

5. AIR TEMPERATURE AND RELATIVE HUMIDITY

To get a simultaneous reading of wet and dry bulb temperatures, a wet and dry bulb thermometer are mounted side by side on a frame fitted by which the two thermometers can be whirled through the air. The whirling is stopped from time to time and readings taken until the temperature reading of the wet bulb thermometer starts to rise, *the lowest reading being taken as the wet bulb thermometer.*



Sling psychrometer with wet bulb and dry bulb stems and handle to whirl the instrument until the wet bulb settles.

A mercury-in-glass thermometer may be needed to check whether air density corrections should be made at the final stage in measuring total volume flow from the fan(s). A high degree of accuracy is unnecessary.

Electric thermometers may also be used and these are usually one of *two* types. One is the thermocouple type, in which the thermocouple generates an electrical current when temperature varies between the hot and cold junction of dissimilar metals. By reading the amount of current generated, an indication of temperature is given.

The other type of electric thermometer is the resistance thermometer, the type generally known as the *Hot Wire Anemometer* (see Page 23) covered elsewhere, since it also is used as an air velocity measuring instrument.

6. VOLTAGE AND CURRENT

It is important to check all motor current at the beginning of balancing to see if it is within design range.

Induction Ammeter

A combined induction ammeter and voltmeter is generally used for current measurement. This does not have to be wired into the circuit; it is easy to use and has an accuracy of 3% full scale. The trigger operated jaws of the ammeter is placed around the conductor at any convenient point and the induced current in the ammeter is indicated as the actual current flow in the conductor. Insulation does not affect the reading, but the instrument will only work on a single conductor - not a twin cable. With three phase motors, take readings on each of the three wires and average the results.

The instrument described can also be used for measuring voltage, although in practice measurements are not often required.

7. SOUND LEVEL

Noise and vibration caused by fans, pumps, compressors, etc., may produce sound. Noise is simply undesirable sound. The frequency range of human hearing varies from person to person and age reduces ability to hear higher frequencies. The *audio-frequency range* for humans is generally 20 Hz to 20,000 Hz.

The audio-frequency range is divided into 10 groups *(octave bands)*. The *centre frequency (geometric mean frequency)* of each band is half that of the next higher band. High frequency will be heard as high-pitched sound.

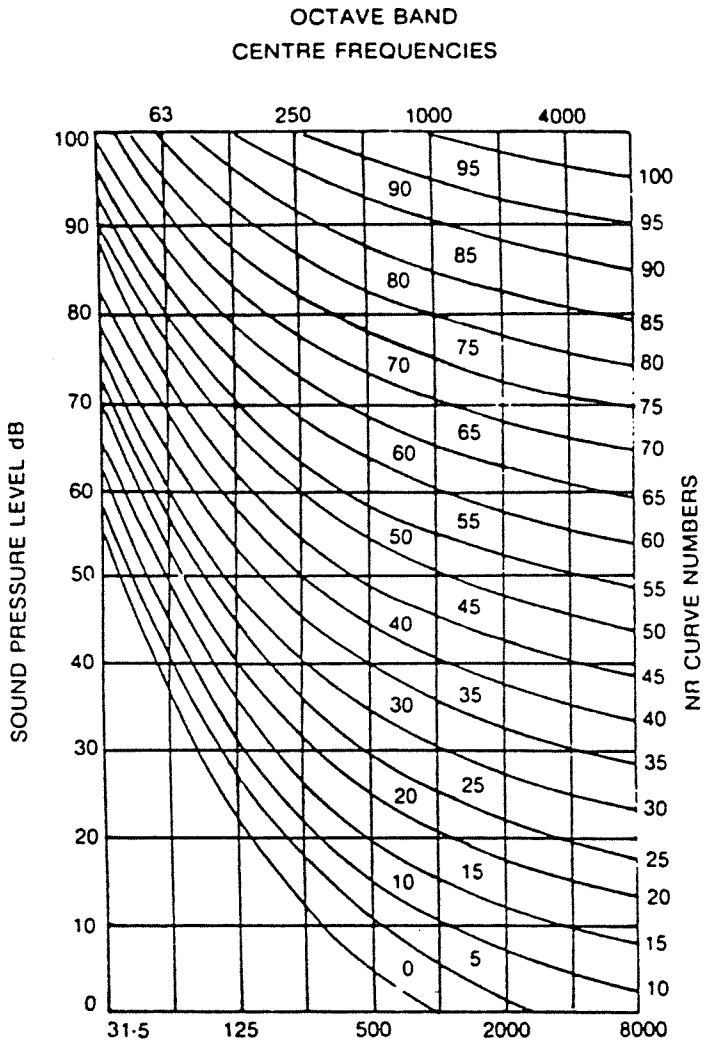
Table

Band number	Frequency limits (Hz)	Centre frequency (Hz)
1	20-50	31·5
2	50-78	63
3	78-200	125
4	200-312	250
5	312-800	500
6	800-1 250	1 000
7	1 250-3 200	2 000
8	3 200-5 000	4 000
9	5 000-12 800	8 000
10	12 800-20 000	16 000

The irritation caused by noise depends upon frequency and loudness. The unit used for sound measurement is the *Decibel (dB)*. The decibel scale is a logarithmic scale which results in a small range of numbers to cover the audible range of the human ear.

It is to be noted that decibels cannot be added or subtracted by ordinary arithmetic.

For each frequency band, noises as experienced by the human ear have been ascertained and the *noise rating (N.R.)* curves established (N.R. curves are in S.I. units) the older N.C. curves are not. (Both curves are interchangeable).



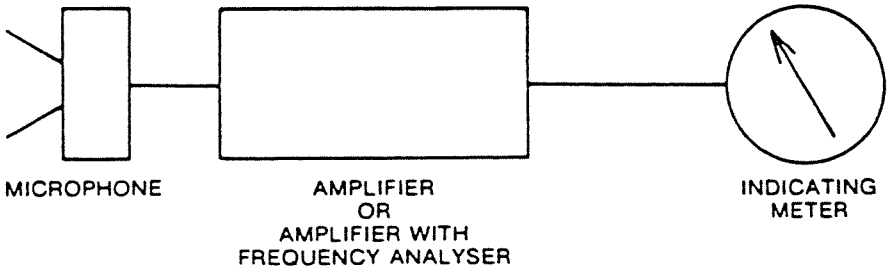
OCTAVE BAND CENTRE FREQUENCIES NR CURVES

The N.R. curves are used to determine the noise present in a situation; N.R. ratings above 85 can cause temporary or permanent hearing damage.

Table
Recommended noise ratings

Recommended noise rating curve	Typical application
NR 20	Concert halls, opera halls, sound studios, large theatres
NR 25	Lecture rooms, small theatres, cathedrals
NR 30	Living rooms, board rooms, offices, conferences, small lecture rooms
NR 35	Hotel rooms, hospital wards, small offices
NR 40	Restaurants, bars, night clubs, departmental stores, lobbies, post offices, reception areas, shops
NR 45	Cafeterias, canteens, supermarkets, swimming pools, bowling clubs, laundry rooms, kitchens, computer rooms, accounting machine rooms
NR 50 and above	Justifiable in factories

To arrive at the noise level, of say a supply air fan, involves *measuring sound pressure level at different frequency bands.* These valves are then used in conjunction with the N.R. curves and recommended noise ratings to determine if the sound level is acceptable. Either a *Sound Level Meter* or a *Frequency Analyzer* can be used. Both instruments work on the same principle.



The microphone picks up and converts the sound impulses into electronic impulses. These are amplified and passed on to the indicating meter which gives sound pressure readings in dB.

The *sound level meter* consists of a microphone, an amplifier and an indicating meter. The *frequency analyzer* consists of a microphone, an amplifier with frequency analyzer and an indicating meter. In the sound level meter, there are three scales; scale A is used for low noise intensity levels (55 dB and below); scale B for medium noise + intensity levels (55 dB to 85 dB); and scale C for high noise (85 dB and above). In the frequency analyzer, the sound pressure levels in each of the octave bands can be obtained. Therefore, the frequency analyzer gives more complete results. These instruments are compact and can be held in the hand while taking readings many being battery operated. The instrument must be held at a height of 1.5m above the floor level and at least 1.0m away from the sources of the noise.

SECTION 1.

WORKSHOP/ASSESSMENT SHEET

1. Manometers and Velometers are both used in air balancing.

   (a) One measures pressure. What does the other instrument measure?

   _____

   _____

   (b) Which instrument measures pressure?

   _____

   _____

   (c) Give the unit or units of measurement for each instrument.

   Manometer: _____

   Velometer: _____

2. Could more information be found by using a *Hot Wire Anemometer* than a *Rotating Vane Anemometer?*
   Fully explain your answer.

   _____

   _____

   _____

   _____

   _____

   _____

3. With which instrument or combination of instruments would you use the following formula?

$$1.3\sqrt{Pv}$$

   _____

   _____

4. If using the formula in Question 3, then in what units of measurement would you be recording?

5. Obtain a Rotating Vane Anemometer from the store and on the register nominated by the teacher, find the average air flow in l/s.
Give a full description of the procedure you followed.

6. The formula: $Pv = Pt - Ps$ is very important because we need to find Pv in duct systems.

(a) Why do we not just measure Pv directly?

(b) Why is Pv so important?

7. Sketch and label how you would find the velocity of air following in a duct using a *Pitot* tube and show all pressures and indicate direction of air flow.

8. When the teacher is satisfied with the answer to Question 7, obtain required instrument or instruments from the store and, where indicated by the teacher, find the average air flow in l/s and m^3/s in a duct. Give a full description of the procedure you adopted.

9. (a) Obtain from the store a *Deflecting Vane Anemometer* and, using the same register as nominated by the teacher in Question 5, find the average air flow in l/s.
Give a full description of the procedure you followed.

(b) Do the readings correspond?

(c) What are the two volumes?

(i) Rotating Vane: _____

(ii) Deflecting Vane: _____

(d) Were the instruments both checked for calibration?

_____

_____

10. (a) A duct 600mm square when connected to a water manometer gives a reading of 12mm.
Find air volume flowing in the duct in l/s and m^3/s.
Show all calculations.

(b) A duct 500mm x 225mm should have a volume flow rate of 944 l/s. What should be the reading in Velocity Head (mm) on a water manometer?
Show all calculations and convert the answer (Hv) into Pv (Pa).

VENTILATION NR 13

Student worksheet 13 4:2

To answer the following questions you will need to obtain from the instructor the NEEB Environmental systems books (3).

Question 1.
Name the two areas where sound tests are required.

_____

_____

Question 2.
Identify three scoures of noise that are common in most offices.

_____

_____

_____

Question 3.
What are some methods that can be used to reduce outdoor noise levels from entering a building. List and explain three (3).

_____

_____

_____

_____

_____

_____

_____

Question 4.

List four (4) types noise that could come from the Air Conditioning and air handling system.

_____

_____

_____

_____

Question 5.
How are vibrations transmitted through a building.

Question 6.
What are the major sources of vibration in Air Conditioning systems. List seven (7) scoures.

Question 7.
How is equal deflection under unequal loads achieved ?

Question 8.
Apart from mechanical equipment,what other types of equipment should have isolators fitted ?

Question 9.
Describe the construction of the open exposed spring type isolator.

Question 10.
With the aid of a diagram explain the operation and possible problems of this type of isolator.

Question 11.
On what of equipment may Neoprene Pads be used ?

Question 12.
What type isolation mount could be used to support pipework and explain it's operation.

Question 13.
With the aid of a diagram explain how ductwork should be connected to a fan plenum.

Question 14.
When fitting vibration eliminators to pipework,under what conditions do you need to use metal braid hose ?

LEARNING OUTCOME 5

Assessment:		Student report, practical exercises.
Performance:	a.	Develop an appropriate checklist with all information, formulas etc. needed to carry out an air balance to a given specification.
	b.	Using the above checklist and all other required equipment, air balance a ventilation/air conditioning system that has several fixed volume supply registers.

AIR BALANCE

- Specifications
- Methods of ratio
- Application of test equipment
- Rules of thumb
- Checklists
- Equipment required
- New installations/existing installations
- Fault Finding
- Adjustment methods
- Pitfalls

Suggested teaching time: 4 hours

DEFINITION OF BASIC TERMINOLOGY

(a) *Volume of Air*

Volume of air flowing in a system is the flow rate produced by the fan, independent of the density of the air, expressed as:

(i) Cubic metres per second handled by a fan at any density (m^3/s).

or (ii) Litres per second handled by a fan at any density (l/s).

(b) *Outlet Velocity*

This is the theoretical velocity of the air as it leaves a fan or air distribution outlet, and is calculated by dividing the air volume in m^3/s by the fan or outlet nett area in m^2 and is expressed as:

Metres per second (m/s)

(c) *Laminar and Turbulent Flow*

When air flows through a duct system at low velocities the particles follow paths free from eddy currents or swirls. The flow is then said to be *Laminar*. As the velocity increases, the characteristics of the flow changes, eddy currents form and the air becomes swirly. This type of flow is known as *Turbulent* flow. Laminar flow produces less friction losses, considerably less system noise, but also extremely poor heat transfer coefficients. A special case of laminar flow called *stratification* which is the result of different densities (mixing of return and outside air) can raise problems in air systems.

Turbulent flow produces higher friction losses, reduced chance of stratification and excellent heat transfer.

In general, turbulent flow is more desirable for the overall system.

(d) *Pressure Classification of Ducts*

Classification of duct systems by pressure and/or velocity is quite arbitrary. The *SHEET METAL* and *AIR CONDITIONING CONTRACTORS NATIONAL ASSOCIATION, INC. (SMACNA)* is developing duct construction standards, established the following breakdown.

(i) Low Velocity

Up to 10 m/s and up to 50mm (500 Pa) static pressure.

(ii) High Velocity

Above 10 m/s.

(a) Medium Pressure: up to 150mm (1500 Pa) static pressure.

(b) High Pressure: Over 150mm (1500 Pa) static pressure. Up to 250mm (2500 Pa) static pressure.

NOTE: Medium and high pressure duct systems should be tested for leaks because performance of systems having such ductwork is affected by air leaks to a much greater extent than low velocity systems.

(e) *Velocity, Static and Total Pressure*

(i) Velocity Pressure *(Pv)* is the pressure which air in a duct exerts due to its motion.

(ii) Static Pressure *(Ps)* is the pressure within a duct which tends to burst the duct.

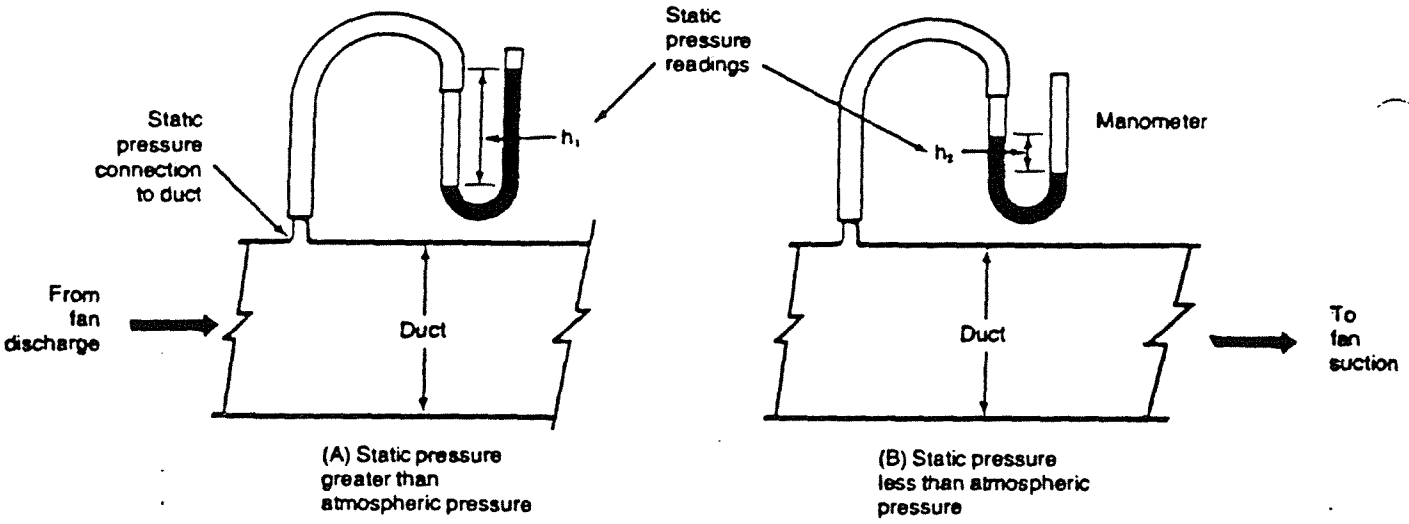
(iii) Total Pressure *(Pt)* is the sum of the static pressure *(Ps)* and the velocity pressure *(Pv)*.

$$Pt = Ps + Pv$$

The static, velocity and total pressures should not be measured in mm of water or any other liquid, as the unit of pressure is the Pascal or Pa. When these pressures are, however, measured in mm of water, they should be termed static, velocity and total *heads*, respectively.
The head in mm of water must be multiplied by 9.81 to obtain the pressure in Pa. (Multiply by 10 is close enough for practical purposes).

1. THE SIGNIFICANCE OF STATIC PRESSURE (Ps)

Static Pressure (Ps) is a measure of the resistance that a duct system presents to the flow of air. Just as a given value of static pressure will push the liquid in a manometer column, as shown on following page, so it has the ability to push air through a duct.

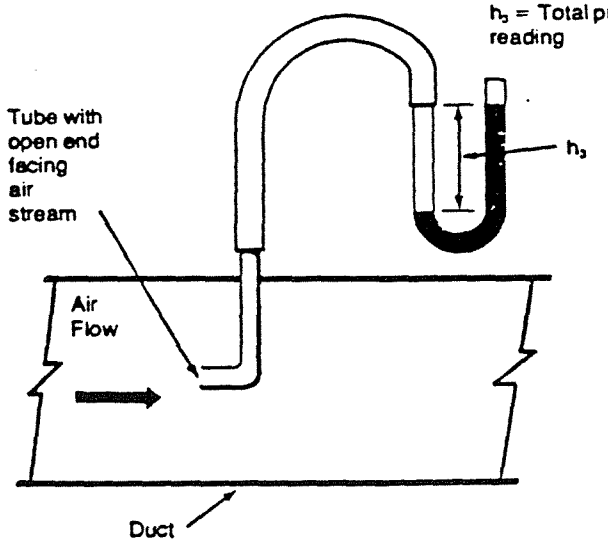


Measuring static pressure

Fans are rated on the basis of the amount of static pressure (Ps) they can develop, as an indication of the amount of duct resistance they can overcome.

A difference in static pressure (Ps) between that at the fan and that at the far end of the duct results in flow of air in the duct.

2. TOTAL PRESSURE (P_T)



Measuring total pressure

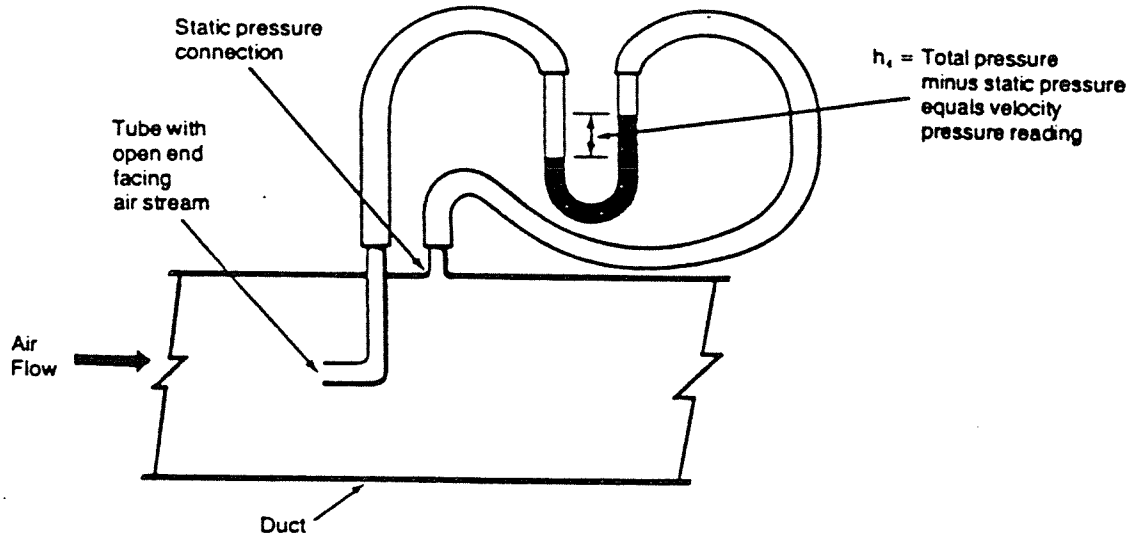
A tube, often called an *impact tube*, is placed so as to face directly into the air stream. In this position, the pressure transmitted to the manometer will be the total of the velocity pressure (Pv) plus the static pressure (Pv) in the duct.

3. THE SIGNIFICANCE OF VELOCITY PRESSURE (Pv)

Velocity Pressure (Pv) is an important figure because it enables us to calculate the volume of air flowing in a duct. Velocity pressure (Pv) cannot be measured directly but is found as the difference between the total pressure (Pt) and the static pressure (Ps).

$$Pv = Pt - Ps$$

This can be done by taking separate pressure readings and then subtracting one reading from the other. Normally, velocity pressure (Pv) is found directly by using the connections shown below.


Static pressure connection
Tube with open end facing air stream
h_t = Total pressure minus static pressure equals velocity pressure reading
Air Flow
Duct

Measuring velocity pressure

The greater the velocity, the greater will be the velocity pressure (Pv). If we measure the velocity pressure (Pv) we can calculate the corresponding velocity as follows:

(a) Assume density of air as 1.2 kg/m^3.

(b) The manometer is graduated in millimetres of water.

$$Pv = \frac{\rho V^2}{2}$$

where: Pv = Velocity Pressure, Pascals, Pa.
ρ = Density kg/m^3.
V = Velocity metres/second. m/s.

$$Pv = \frac{1.2 \times V^2}{2}$$

$$Pv = 0.6\ V^2$$

$$V^2 = \frac{Pv}{0.6}$$

$$V = \sqrt{\frac{Pv}{0.6}}$$

$$V = \sqrt{\frac{Hv \times 9.81}{0.6}}$$ (Hv = Velocity head in mm of water)

$$V = 4.04\ \sqrt{Hv}$$

OR

(a) Assume density of air as 1.2 kg/m^3.

(b) The manometer is graduated in Pascals (Pa).

$$Pv = \frac{\rho V^2}{2}$$

where: Pv = Velocity pressure, Pascals, Pa.
ρ = Density kg/m^3.
V = Velocity metres/second, m/s.

$$Pv = \frac{1.2 \times V^2}{2}$$

$$Pv = 0.6\ V^2$$

$$V^2 = \frac{Pv}{0.6}$$

$$V = \sqrt{\frac{Pv}{0.6}}$$

$$V = 1.3\ \sqrt{Pv}$$

These formulas are essential in testing and balancing because with instruments and procedures covered later, the velocity pressure (Pv) in a duct can be measured and then the velocity can be calculated using these formulas.

4. Air velocity in a duct determines the volume of air flowing in the duct as found by:

$$Volume\ (m^3/s) = Velocity\ (m/s) \times Area\ (m^2)$$

5. Duct sizes are given in millimetres, therefore, *Area.*

(a) For a round duct

$$Area\ (m^2) = \frac{\Pi D^2}{4} \quad \underline{or} \quad = \Pi r^2$$

where: D = Diameter of duct in millimetres
r = Radius of duct in millimetres

$$Area\ (m^2) = \frac{\Pi \times \left(\frac{Dmm}{1000}\right)^2}{4} \quad \underline{or} \quad \Pi \times \left(\frac{rmm}{1000}\right)^2$$

(b) For a square or rectangular duct

$$Area\ (m^2) = Length\ (L) \times Depth\ (d)$$

$$Area\ (m^2) = \left(\frac{Lmm}{1000}\right) \times \left(\frac{dmm}{1000}\right)$$

EXAMPLES

1. Duct size = 450mm x 350mm
Velocity = 7.5 m/s
What is the volume in m^3/s?

Solution

Volume (m^3/s) = Velocity (m/s) x Area (m^2)

$$V = 7.5 \times \left(\frac{450}{1000}\right) \times \left(\frac{350}{1000}\right)$$

$$V = \underline{1.18}\ m^3/s \text{ (approx.)}$$

2. Specified Air flow = 1.0 m^3/s
Duct size = 375mm diameter (round duct)
What should the velocity be at the specified air flow?

Solution

Volume (m^3/s) = Velocity (m/s) x Area (m^2)

$\therefore$ Velocity (m/s) = $\frac{\text{Volume } (m^3/s)}{\text{Area } (m^2)}$

Area (m^2) for round duct = $\frac{\Pi \left(\frac{D}{(1000)}\right)^2}{4}$

Area (m^2) = $\frac{\Pi \times \left(\frac{375}{(1000)}\right)^2}{4}$

Area (m^2) = 0.11 m^2 (approx.)

Velocity (m/s) = $\frac{1.0}{0.11}$

Velocity (m/s) = 9.1 (approx.)

3. Specified Air flow = 1.8 m^3/s
Maximum duct velocity = 10 m^3/s
What is the minimum length of a rectangular duct if depth is to be 300mm?

Solution

Volume (m^3/s) = Velocity (m/s) x Area (m^2)

$\therefore$ Area (m^2) = $\frac{\text{Volume } (m^3/s)}{\text{Velocity (m/s)}}$

Area = $\frac{1.8 \ (m^3/s)}{10 \ (m/s)}$

Area = 0.18 m^2

Area (m^2) = $\frac{(\text{Length})}{(\ 1000\ )} \times \frac{(\text{width})}{(1000\ )}$

$\therefore$ $\frac{(\text{Length})}{(\ 1000\ )} = \frac{\text{Area } (m^2)}{\frac{(\text{widthmm})}{(\ 1000\ )}}$

The various techniques for measuring the air flow in duct systems and at terminal devices are controversial, and none is universally accepted. It is difficult to measure air velocities and flow rates in the field. All methods are subjected to the ability of the balancer. Proper balancing is time consuming and requires expertize and diligence. At present, no one established procedure can be considered as applicable to all systems. However, there is one point of agreement: *air systems should be balanced before the hydronic, steam, and refrigerant systems.*

1. The *minimum* instruments necessary for air balancing are:

   (a) Incline manometer calibrated in no less than 1mm divisions (10 Pa).

   (b) Combination inclined and vertical manometer (0-250mm (0-2500 Pa) is generally the most useful).

   (c) Pitot tubes usually 450mm and 1200mm long tubes cover most balancing requirements.

   (d) A tachometer which should be of the high quality, direct contact, self-timing type.

   (e) Clamp-on ampere meter with voltage scales.

   (f) Deflecting vane anemometer.

   (g) Rotating vane anemometer.

   (h) Thermal-type (hot wire) anemometer.

   (i) Dial and glass stem thermometers.

2. Before beginning to balance the system, eliminate every possible air flow restriction. Open all air valves, fire dampers, and volume controls in both supply and return ducts. Adjust outside air dampers for minimum and maximum positions and adjust return air dampers for maximum flow. Set adjustable pattern ceiling diffusers for horizontal air discharge patterns, whenever possible.

3. Before any system can be balanced properly, the supply fan must develop enough static pressure (Ps) for the system, and the air volume handled by the fan must be adequate for the system. Therefore, after ensuring that all related fans (supply, return, exhaust) are operating, measure and compare with specifications.

   (a) System Static Pressure.

   (b) Fan RPM, voltage at fan motor and current drawn.

   (c) Total air volume.

(A) Contrary to what may be supposed, fan static pressure is *not* simply the difference between outlet static pressure and inlet static pressure. By definition, fan static pressure is equal to the rise in total pressure across the fan minus the velocity pressure created by the fan.

Fan (Ps) (static pressure) = Fan (Pto) (outlet total pressure) - Fan (Pti) (inlet total pressure) - Fan (Pvo) (outlet velocity pressure)

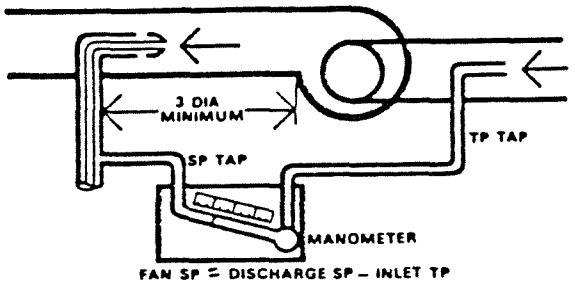
Since Pt = Pv + Ps, Outlet Pt = Outlet Ps + Outlet Pv and inserting this for Outlet Pt in the above formula, we have:

Fan Ps = (Outlet Ps + Outlet Pv) - Inlet Pt - Outlet Pv

and as *Outlet Pv* cancel out (negative and positive), we are left with:

Fan Ps = Fan Pso (outlet) - Fan Pti (inlet)

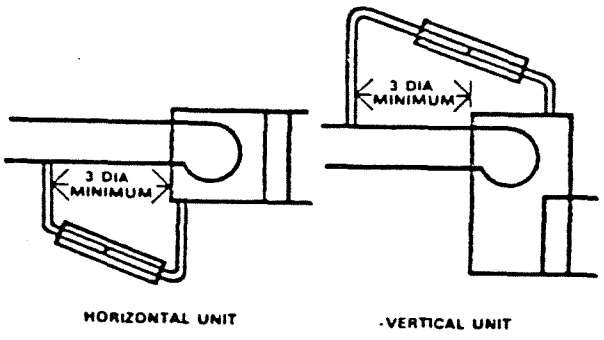
To measure System Static Pressure, which is also the Fan Static Pressure, either two separate readings or two pitot tubes must be used as shown opposite when ducted fans are installed.



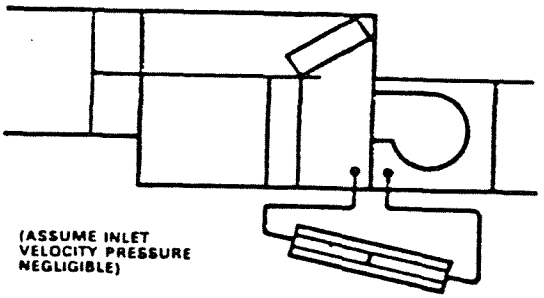
Fan Static Pressure—Ducted Fans

Where fans are installed in plenum chambers inlet velocity pressure is not measurable and is taken as zero. *Therefore, Inlet Total Pressure is equal to Inlet Static Pressure* and

Fan Ps = Fan Pso (outlet) - Fan Psi (inlet)



Static Pressure Measurements—
Draw-Thru Air Handling Units



Fan Static Pressure With Non-Ducted Fans

(B) To measure fan R.P.M. use a tachometer. Hold the tachometer in contact with the fan shaft, make several readings, average the measurements and record the average reading.

The balancer is checking the rotation speed of the fan.

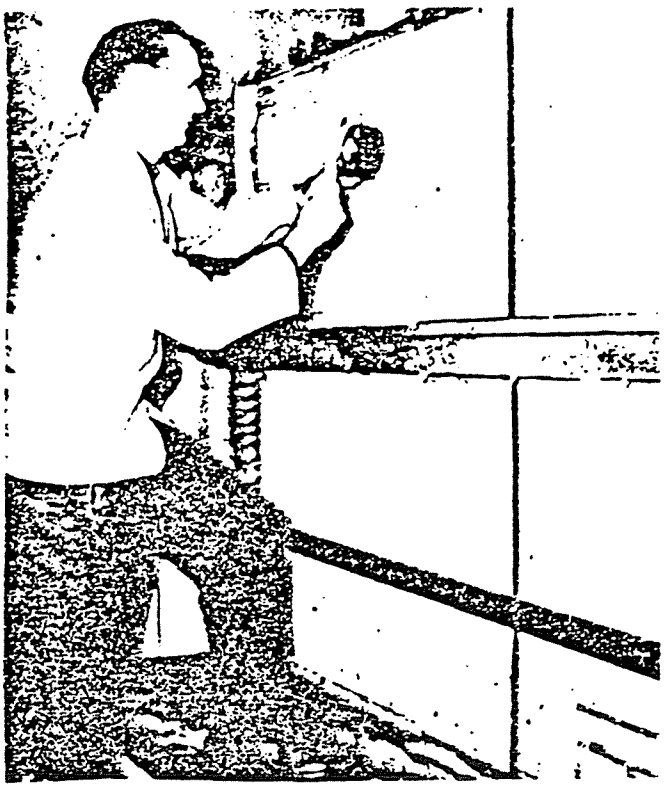
To measure fan motor voltage and current drawn use a volt-amp meter. Connect the volt-amp meter to the power input terminals to read voltage. Record the readings. Connect the meter in the power line and read current drawn. Record the reading.

Compare the fan R.P.M. voltage and current drawn with those on the motor nameplate. Readings should not exceed the nameplate ratings of the motor.

(C) To measure total air flow, use a *Velometer* or stopwatch and *Rotating Anemometer*. Three alternate methods are commonly used.

(i) Velocity traverse across the cooling or heating coils

For best results, when using the Anemometer and stopwatch measurements must be made on the downstream side of the coil. Position the Anemometer about 25mm from the coil surface. Before beginning, divide the coil into equal sections and calculate the area of each section in m^2. Multiply the velocity in m/s by the area in m^2 for each section and total all section volumes to arrive at the *Total Volume*.



The balancer is taking a reading across the coils to determine velocity using a rotating vane anemometer.

As shown opposite, a Velometer can also be used to measure velocity and, therefore, calculate total air volume.

(ii) Traverse across filters

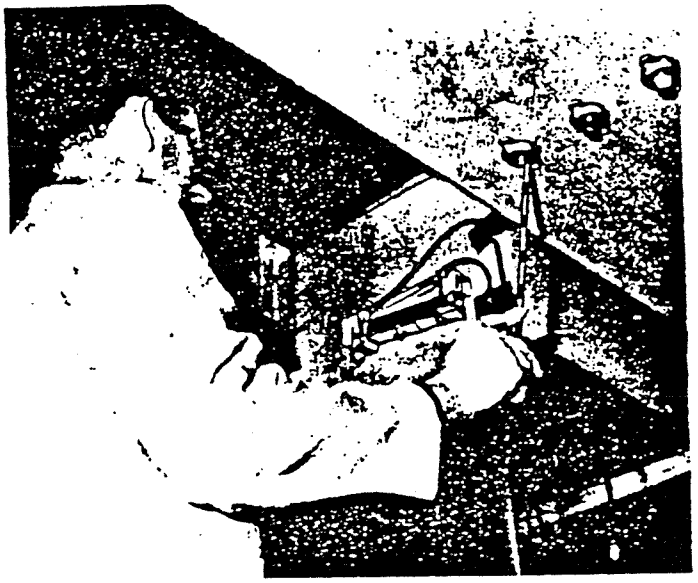
Sometimes we must use an alternate location for measuring total air volume. The same methods covered previously can be used to measure total air volume at the filter location.

The balancer is taking a reading across the filter to determine velocity using a velometer.

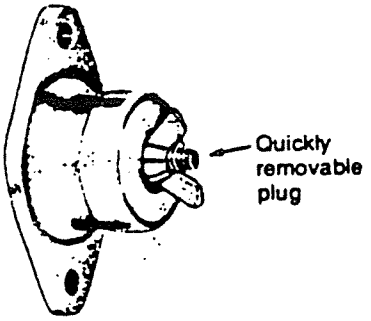
(iii) Traverse in main duct

A third alternative for measuring total air volume is in the main discharge duct. Measurements *must* be made in a *straight section of duct* approximately *ten duct diameters downstream from an elbow or fan and, between two and five duct diameters upstream from an elbow on take off.*

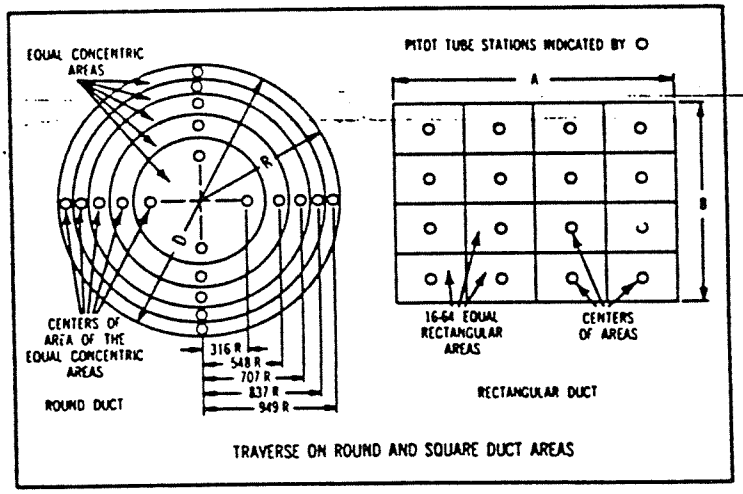
A Velometer fitted with a duct jet may be used to traverse the duct, and this is the preferred instrument when the duct velocity is *below* 5 m/s.



When the duct velocity is *above* 5 m/s then the *pitot* tube is generally used to calculate the total air flow.



Instrument test port



TRAVERSE ON ROUND AND SQUARE DUCT AREAS

Regardless of the instrument used the drawing opposite illustrates the method of arriving at sensing points across the duct cross section.

For traversing square or rectangular ducts at least 16 and up to 64 are required depending on duct size. For traverses of less than 64 sensing points, the minimum distance between centres should not exceed 150mm.

All the material covered in Section 11 discusses preliminary air balancing and must be fully documented. Final air balancing depends on the kind of system to be balanced.

A sample of report sheets are shown on the following pages.
If supply fan volume is not within plus or minus 10 percent of design volume, adjust fan speed to obtain approximate design air volume.

AIR BALANCING WORK SHEET

Air Moving Equipment

Barber-Colman Company
Air Distribution Division
1300 Rock Street
Rockford, IL 61101

Date _____ Test No. _____

Job Name _____

_____

Location _____

_____

System _____

_____

Description				
Mfg.				
Model No.				
Serial No.				
Operating Conditions	Specified	Actual	Specified	Actual
Total Volume				
R. A. Volume				
O. A. Volume				
Suction S. P.				
Discharge S. P.				
Total S. P.				
rpm				
Motor Mfg.				
KW Input				
Voltage				
Amperage				

By _____

BARBER COLMAN

AIR BALANCING WORK SHEET

Filter/Coil Velocity Traverse

Filter/Coil Size _____ Date _____ Test No. _____

Job Name _____

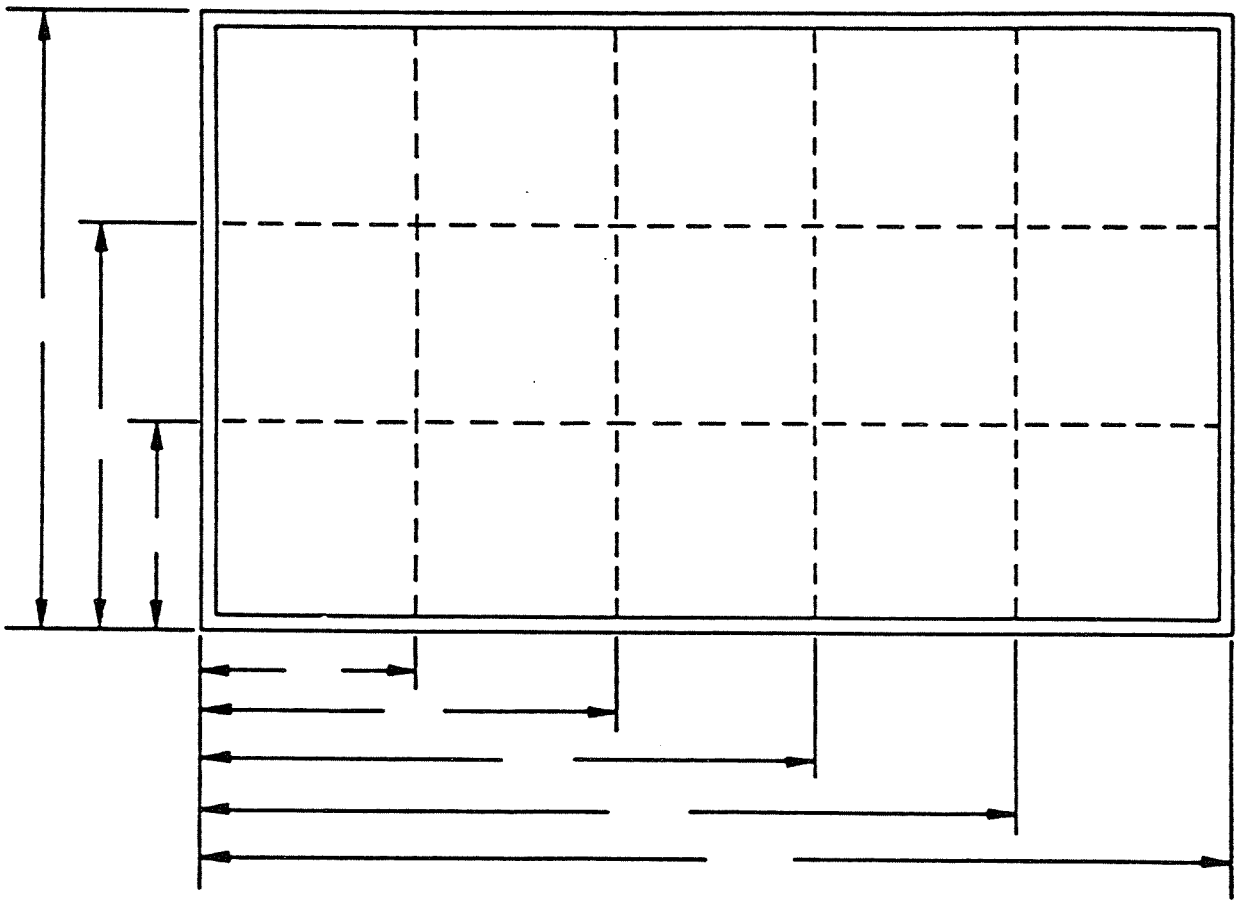
Filter/Coil Area _____

Location _____

Design m^3/s _____

System _____

Actual m^3/s _____



Area (sq. m.) X Average Velocity (m/s) = Air Volume (m^3/s)

_____ sq. m X _____ m/s . = _____ m^3/s

Area — Average Velocity — Air Volume

Instrument _____

By _____

AIR BALANCING WORK SHEET

Duct Velocity Traverse

Barber-Colman Company
Air Distribution Division
1300 Rock Street
Rockford, IL 61101

Date_____ Test No._____

Job Name_____

Location_____

System_____

Duct Size_____

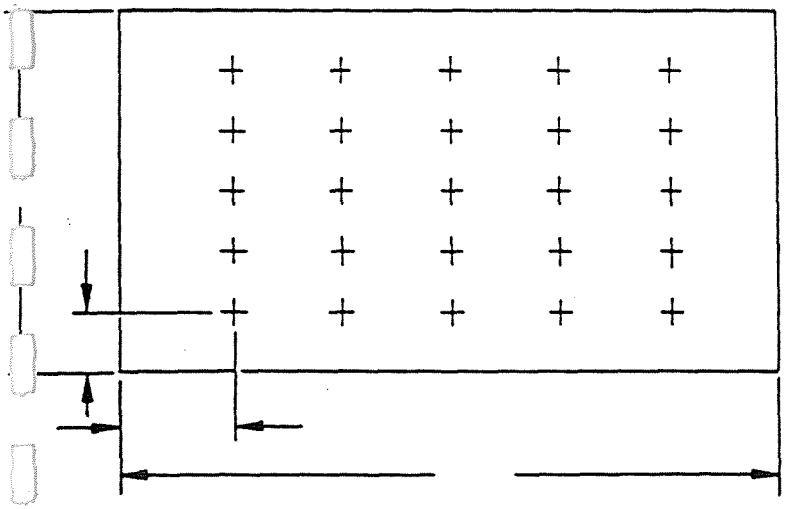
Duct Area_____

Design m^3/s_____

Actual m^3/s_____

Traverse Points	DUCT VELOCITIES - m/s			
	Traverse			
	1	2	3	4
1				
2				
3				
4				
5				
6				
7				
8				
9				
10				
11				
12				
13				
14				
15				
16				
17				
18				
19				
20				
21				
22				
23				
24				
25				
Average Velocity				
Duct Area				
Total Volume				

Layout



Instrument

By

4. Determining the correct proportion of outside to total air is basic to the proper balancing of any system. The most practical method of setting outside air proportional dampers is the *mixed air temperature* method.

$$Tmix = (\%\ OA \times Toa) + (\%\ RA \times Tra)$$

where: Tmix = Temperature of air mixture
Toa = Temperature of outdoor air
Tra = Temperature of return air
%OA = Percentage of outdoor air to total air
%RA = Percentage of return air to total air

The formula may be restated as:

$$\%\ OA = \frac{Tra - Tmix}{Tra - Toa}$$

Example: A system designed for 7.0 m^3/s total air. The minimum setting for the outside air damper calls for 1.1 m^3/s. Determine the correct setting of the outdoor air damper.

STEP 1. Using the pitot tube traverse at the proper point of the supply duct, set the fan speed to deliver 7 m^3/s.

STEP 2. Insert a set of calibrated thermometers at strategic points to measure the temperature of the outdoor air, return air and mixed air. Assume that these read 33°C outside, 24°C return air and 24.5°C mixture.

STEP 3. Find the design percentage of outdoor air.

$$\frac{1.1}{7.0} = 0.157$$

STEP 4. Calculate the air mixture temperature required to give 15.7% outdoor air by using the formula.

$$Tmix = (0.157 \times 33) + (0.843 \times 24) = 25.4^{\circ}C$$

STEP 5. Since the recorded mixture temperature is 24.5°C and the required mixture temperature is 25.4°C, the system is obviously short of outdoor air (by formula 5.6%), therefore, open the outdoor damper slightly to allow more outdoor air and close the return air damper slightly to decrease return air keeping the supply volume at 7.0 m^3/s until the air mixture temperature rises to 25.4°C.

PROBLEM

Supply air quantity = 4.72 m^3/s.
Specified minimum outside air quantity = 940 l/s.
From actual measurements, Toa = 32°C
Tra = 21°C and Tmix = 24°C.

Find:

(i) The temperature of the mixed air stream that will result from the correct outside air - return air ratio.

(ii) Discuss correction procedures if they are required.
Show all calculations.

5. Balancing Procedures

AJUST ALL REGISTERS IN THE SYSTEM TO WIDE OPEN.

(a) One suggested procedure is to first check the furthest outlet in each branch. If the outlet is below design velocity, leave the damper fully open and move onto the next upstream outlet. If the outlet velocity is above design, then throttle before moving to the next upstream outlet.

It is important that the balancer refers to the manufacturer's data for the proper "K" factor to use in conjunction with the instrument used. It is also important to note that the manufacturer will designate various locations for taking readings on different models of outlets.

DIFFUSER WIDTH (mm)	DIFFUSER HEIGHT (mm)								
	75	100	125	150	200	250	300	350	400
150	.06	.10	.13	.17					
200	.09	.13	.18	.23	.33				
250	.12	.18	.24	.30	.42	.55			
300	.14	.21	.29	.37	.52	.67	.82		
350	.17	.26	.34	.43	.61	.79			
400	.19	.30	.40	.50	.71				
450	.22	.33	.45	.57					
500	.24	.38	.51						
550	.27	.42							
600	.30								
650	.33								
700									

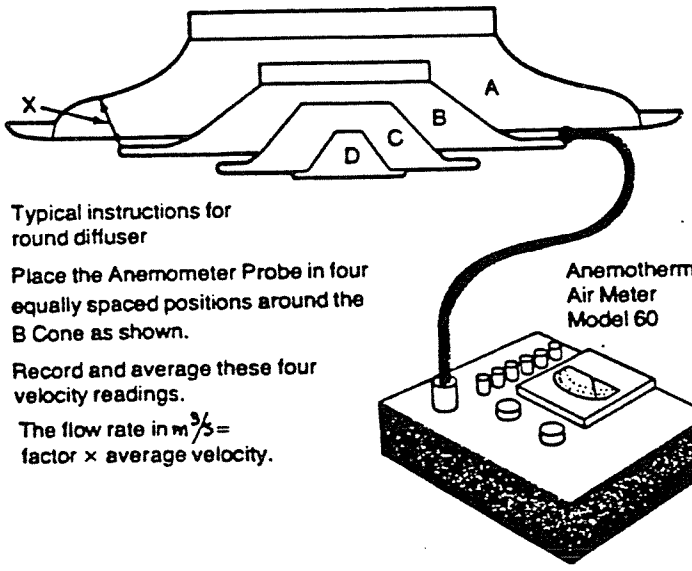
Portion of a table showing net core area of grilles (side wall diffusers)

Example of a table of diffuser factors

Determine the average face velocity from the corrected Anemometer reading.
Determine net core area from Table.
Determine air volume as follows:

Average face velocity in m/s × net core area in sq. m × diffuser factor = volume in m^3/s

Corrected Anemometer Velocity Reading in m/s	1	1.5	2	2.5	3	4	5
Factor	.72	.75	.77	.80	.83	.88	.93



Typical instructions for round diffuser

Place the Anemometer Probe in four equally spaced positions around the B Cone as shown.

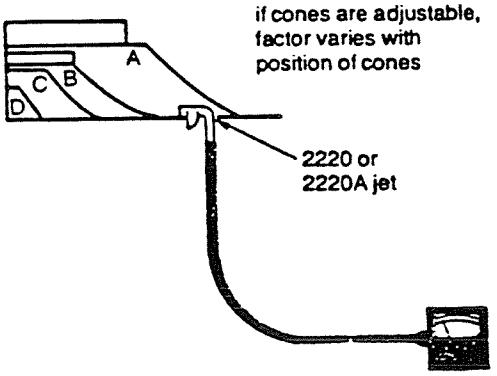
Record and average these four velocity readings.

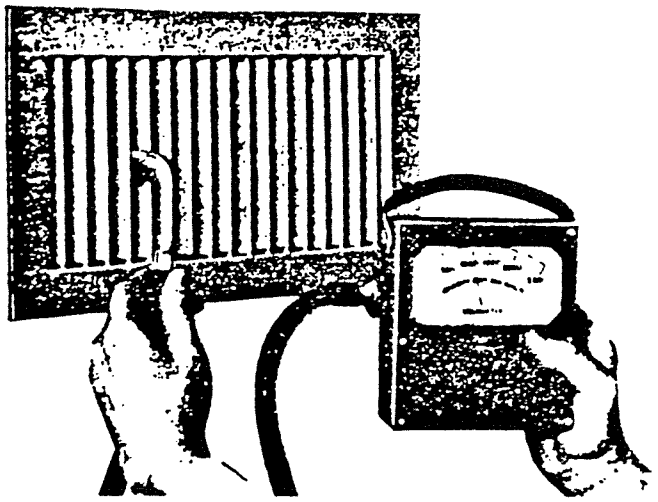
The flow rate in m^3/s = factor × average velocity.

Anemotherm Air Meter Model 60

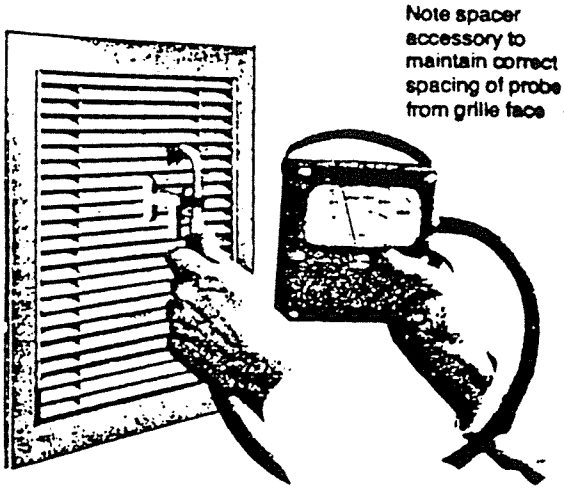
Neck Diam. mm	Cones X	Up Factor	Cones X	Down Factor
150	16	0.18	33	0.22
200	21	0.32	45	0.38
250	25	0.50	55	0.60
300	30	0.72		
350	35			

Portion of a table of flow factors. Note that if cones are adjustable, factor varies with position of cones



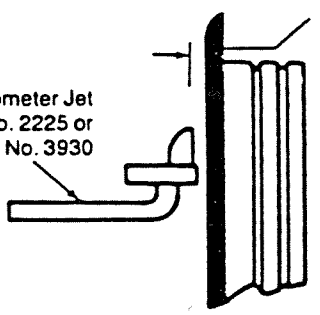


Measuring flow from supply grille with deflecting vane anemometer

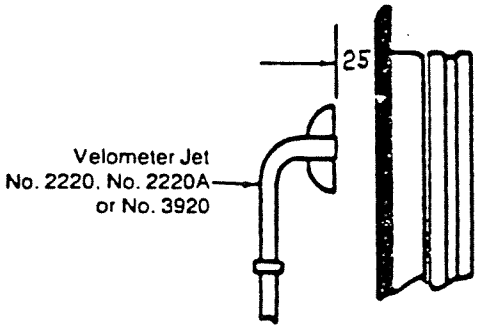


Measuring flow into return or exhaust grille with deflecting vane anemometer

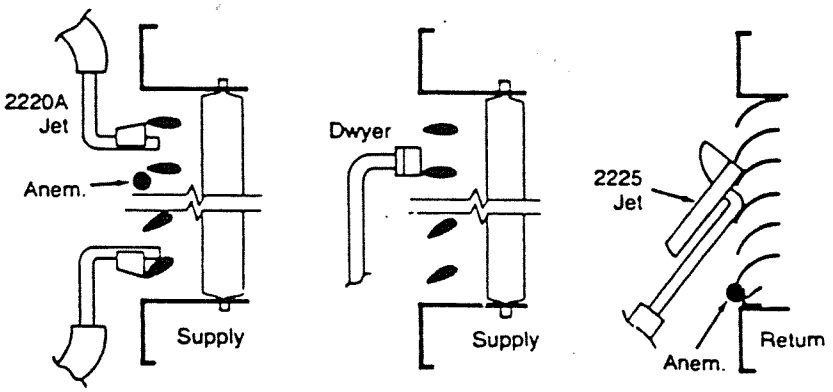
Determine net core area from Table.
Determine air volume as follows: Average face velocity in m/s × net core area in sq. m. = volume in m^3/s



Placing the probe when air flows into grille



Correct placement of anemometer probe for one type of grille



Probe locations for several instrument types

SUPPLY OUTLETS						
CORES	ACCESSORY	DEFLECTION	ALNOR*	ANEM	RVA	DWYER
271	NONE	0	.83	.90	.82	.89
		22½	.80	.88	.81	.86
272		45	.68	.83	.77	.68
"	4 & 5 (Dampers)	0	.78	.88	.80	
		22½	.75			
		45				
281	NONE	0				
		22½				
		45				

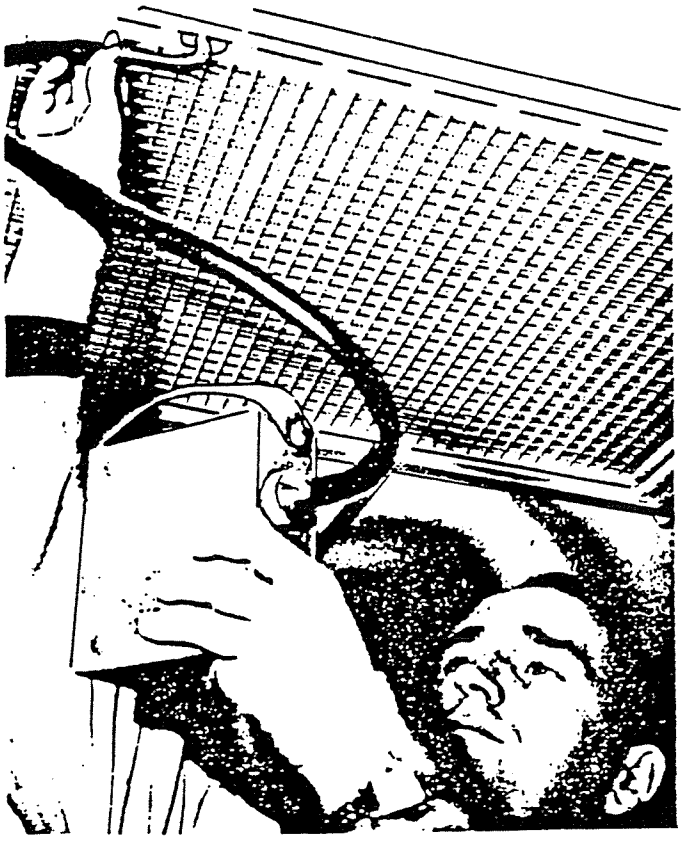
Examples of flow factor tables

RETURN INLET				
CORE	DEFL	ALNOR	ANEM	RVA
23	45°	.45	.50	.65
13	0°	.92	.73	.80
30	0°	.91	75	.80
T-700		.62	.60	71
T-800, T-8000				
25				

Flow rate in m^3/s =
Factor × Core area × Average velocity

Measuring flow through grilles using deflecting vane anemometer and other instruments that give spot readings

As the adjustment of one outlet affects the others, it will be necessary to repeat the above procedure one or more times making finer adjustments each time.



The velometer is being used to take a reading at a light troffer. Note location of the probe.

This sidewall grille is being checked for m^3/s delivery with an anemometer.

(b) A second procedure often used is referred to as the *Balancing from fan out method.* This method is similar to that previously described and begins by each branch being adjusted (using splitter or manual volume dampers) until an approximate balance between branches is achieved.

This branch balance must also be reached before proceeding with the individual outlet correction previously discussed in Section 5(a).

Proceed with balancing individual outlets. Begin by measuring the flow through the outlet nearest the supply fan. Reduce excessive flow by partially closing the volume control damper at the outlet. Move to next downstream outlet from the supply fan and repeat as per outlet number one. Record the measurements on a suitable form.

SYSTEM NO. _____

NO.	SIZE	FACTOR	DESIGN l/s	DESIGN VEL. OR ~~S.P.~~	TEST NO.1 VEL. OR ~~S.P.~~	TEST NO. 2 VEL. OR ~~S.P.~~	TEST NO. 3 VEL OR S P	ACTUAL FINAL VEL OR ~~S.P.~~	ACTUAL FINAL l/s	REMARKS
1	250	.478	200	4.25	4.37	4.22		4.17	196	C D
2	250	.478	200	4.25	4.24	4.17		4.17	196	C D
3	250	.478	200	4.25	4.04	4.17		4.19	197	C D
4	400 x 150	.44	100	2.3	3.05	2.29		2.39	103	TR-E-SET
5	400 x 150	.44	100	2.3	2.77	2.03		2.29	99	TR-E-SET
6	400 x 150	.44	100	2.3	2.35	1.93		2.13	93	TR-E-SET
7	400 x 150	.44	100	2.3	1.99	2.54		2.49	107	TR-E SET
8	400 x 150	.44	100	2.3	1.57	2.39				
9	400 x 150	.44	100	2.3						

Typical outlet report on system balancing

Note that in the form, the column is filled in to show the velocity corresponding to design volume. Therefore, readings in m/s are entered, and adjustment is determined by whether measured velocity is greater or less than 10 per cent of design velocity. Continue to balance progressively upstream from the supply fan until all outlets have been adjusted. Make one or more additional passes until an acceptable balance is achieved.

If all outlets on one branch are high on air flow, it may be required to install an additional volume damper in order to avoid excessive noise generated at the outlets by closing the outlet damper down too much.

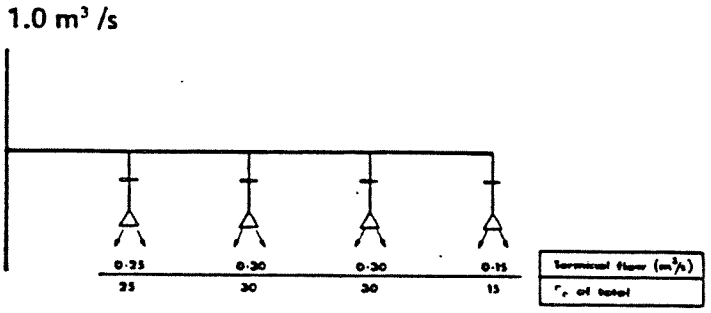
When a satisfactory balance is obtained, calculate and tabulate the actual final volume. Compare to system design volume and make any adjustments as indicated by results tabulated.

(c) The technique of *Proportional Balancing* is generally recognized as the simplest way to regulate an air distribution system. Its greatest advantage is, once a damper has been set, *it never needs to be altered.*

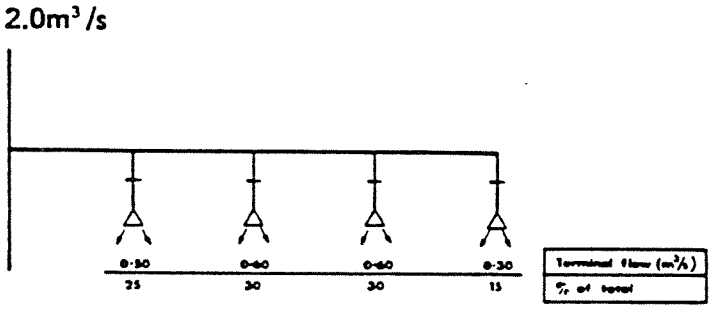
It is not necessary to work with actual design air flow rates to balance outlets or branches, and only one final in-duct measurement of total air flow is needed at the supply fan at the end of the balancing process.

Consider the branch duct shown below. The volume of air delivered by each outlet represents a percentage of the total flow in the duct. Unless the outlet dampers are altered the percentage *(proportion)* will remain the same whatever the flow rate in the duct.

(a) Sub-branch flow



(b) Sub-branch flow



Basis of proportional balancing
Although the flow to the system is altered, the percentage share of each terminal remains the same.

At the present time, there are *two* procedures used for balancing the outlets in the *proportional* or *ratio* method.

(i) The farthest outlet on the farthest branch from the supply fan will be Outlet No.1 on Branch No.1. Number all outlets consecutively, working back towards the fan. Number the outlets similarly on the test report sheet. It is important that the outlets be numbered, tested and adjusted in this sequence. If the sequence is not followed the procedure *will not be valid.*

Determine that Outlet No.1 on Branch No.1 has enough air being delivered to give measurable readings. If not, adjust fan speed, etc., as required. Measure velocities at Outlet No.1 on Branch No.1 (the outlet farthest from fan). Determine average velocities at outlet No.1 and tabulate.

In the following, the designation Vm will mean measured velocity and Vd will indicate design velocity. For outlet No.1, calculate ratio of measured to design velocity as:

$$R = \frac{Vm}{Vd}$$

Call this Ratio R_1 for Outlet No.1

EXAMPLE:

$$Vm_1 = 3.0 \text{ m/s}$$
$$Vd_1 = 2.5 \text{ m/s}$$

$$R_1 = \frac{Vm}{Vd_1}$$

$$R_1 = \frac{3.0}{2.5}$$

$$R_1 = 1.2$$

Proceed to Outlet No.2. Measure flow and determine average velocity Vm for this outlet.

Calculate Ratio R_2 for Outlet No.2.

$$Vm_2 = 3.0 \text{ m/s}$$
$$Vd_2 = 4.1 \text{ m/s}$$

$$\therefore R_2 = \frac{Vm_2}{Vd_2}$$

$$R_2 = \frac{3.0}{4.1}$$

$$R_2 = 0.732$$

Compare R_1 and R_2. If the ratios are not within 10 per cent, adjust outlet No.2 to bring ratios into closer agreement. *Do not adjust Outlet No.1.*

$$R_1 = 1.2 \begin{matrix} + 10\% \\ - 10\% \end{matrix} = 1.32 - 1.08$$

and $$R_2 = 0.732 \begin{matrix} + 10\% \\ - 10\% \end{matrix} = 0.805 - 0.69$$

Measure Vm for both outlets and calculate new values for R_1 and R_2. Continue re-adjusting Outlet No.2, re-measuring and calculating R_1 and R_2 until R_1 and R_2 are within 10 percent of each other and record.

Although neither of the two outlets may have design velocity (or volume), they are now proportionately balanced one to the other.

Proceed to Outlet No.3. Measure and determine average velocity Vm for Outlet No.3 and calculate Ratio R_3 for this outlet. If necessary, adjust Outlet No.3 to bring R_3 within 10 percent of R_2. *Do not adjust Outlets Nos. 1 and 2.* (Adjust of Outlet No.3 automatically changes the ratios of Outlets 2 and 1. The ratio for these outlets approaches the same values. For this reason, once the outlet has been adjusted correctly, it never requires further adjustment.

Proceed to Outlet No.4 and adjust to obtain agreement between R_4 and R_3 within 10 percent.

After all outlets on Branch No.1 are proportionally balanced to each other, proceed to Branch No.2, etc.

(ii) The alternative proportional balance differs as follows: Start regulating outlets on a branch which has a high percentage of design flow after adjusting supply fan for design flow rate (branch could be the closest or farthest or anywhere in between from the fan). Flow rate should not be above 30% of the design figure. Use branch dampers to correct if required. Branches with less than 70% of design flow are left until the high percentage outlets are regulated. This will force air into the low flow rate outlets.

On the selected branch, locate the outlet with the *least percentage* of the design rate (the lowest value of *measured flow/design flow*). Generally, it is the last outlet in the branch. If not, adjust damper in the last terminal until it is working with the same percentage ratio as the one previously measured.

The last terminal is then used as an index against which the ratios from other outlets in the group are compared.

Measure the flow from the outlet next to the index and calculate the percentage ratio. Adjust damper to bring percentage ratio within required tolerance of index outlet. Repeat procedure for next downstream outlet, again comparing it with the index outlet. When all the outlets have been balanced on a branch, each outlet will be running with *an equal percentage* of the design flow (within the allowable tolerances).

To balance the branches, select an outlet well within the tolerance limits (it does not have to be the index terminal used in the outlet balance) for each branch. Find the branch with the least air when comparing *measured flow/design flow.* If it is not the branch farthest from the fan, then adjust branch dampers until it becomes the least favoured and, therefore, the index to which other branches are referred. Starting with the branch next to last, compare percentage of design flow between reference outlets in this branch and the reference outlet in the branch. Adjust duct dampers until the two percentage figures agree within the tolerances (usually $\pm$ 5% for branches). Repeat the procedure with the next upstream branch, again comparing the flow at the selected branch reference outlet with the reference outlet in the index branch.

The balancing of the branches in the technique described in *Section C(i)* differs in that each branch is proportionately balanced to the preceeding branch, and not to an index branch as per *Section C(ii)*, as is as follows:

Select a typical outlet on Branch Nos. 1 and 2 (branches farthest from supply fan). Calculate R ratios for each outlet. Adjust branch splitter or volume dampers until the selected outlets are within 5 percent of each other. The two branch ducts are now proportionately balanced. Proceed to obtain proportionate balance between branches 2 and 3, 3 and 4, etc.

Regardless of which proportioning system is used always work towards the fan, from the end outlet of a sub-branch, the end sub-branch duct, and the end branch on a main duct.

When all branches, etc., are proportionately balanced, check the total air flow at the fan. Adjust fan speed or fan dampers to obtain design flow. The ratio R (Vm/Vd) (Qm/Qd) at the fan will now be 1.0, since the system will be approximately 1.0, and the flow from each outlet will be design air flow rate.

(d) *The basic steps outlined form the foundation for balancing any system. There are, however, a number of variations to the conventional system (i.e. Dual Duct, Multi-Zone, Variable Volume and Induction Systems, etc.) and these, where applicable, will be covered by individial worksheets.*

(e) Service to overcome air supply variations are often required because of the following faults and can be corrected as indicated.

(i) *Dirty filters or coils*

Indication:

_____

_____

_____

_____

Remedy:

_____

_____

_____

_____

(ii) *Fan speed too low*

Indication:

_____

_____

_____

_____

Remedy:

_____

_____

_____

_____

(iii) Belt slipping

Indication:

Remedy:

(iv) Dampers or outlets closed

Indication:

Remedy:

(v) *Restricted ductwork*

Indication:

Remedy:

(vi) *Abnormal temperature drop across the cooling coil*

Indication:

Remedy:

(vii) *High humidity due to warmer coil*

Indication:

Remedy:

(viii) *Draughts due to overblow of outlets*

Indication:

Remedy:

END OF SECTION II.

Additional and more detailed information pertaining to *Section II* will be found in the following:

A.S.H.R.A.E. SYSTEMS VOLUME (1980) CHAPTER 40

A.S.H.R.A.E., 345 East 47th Street, New York, N.Y. 10017.

TESTING, BALANCING AND ADJUSTING OF ENVIRONMENTAL SYSTEMS

William G. Eads, P.E., SMACCNA, 8224 Old Courthouse Road, Tysons Corner, Vienna, Virginia. 22180.

AIR CONDITIONING TESTING AND BALANCING

John Gladstone, Van Nostrand Reinhold Company, New York/Cincinnati/ Toronto/London/Melbourne.

MANUAL FOR THE BALANCING AND ADJUSTMENT OF AIR DISTRIBUTION SYSTEMS (First Edition - 1967)

SMAACCNA, P.O. Box 3506, Washington, D.C. 20007.

START, TEST AND BALANCE - MECHANICAL EQUIPMENT SERVICE MANUAL (First Edition - 1976)

NJS - PAC, Printed in the United States of America.

MANUAL FOR REGULATING AIR CONDITIONING INSTALLATIONS Application Guide 1/75

B.S.R.I.A., Old Bracknell Lane, Bracknell, Berkshire, RG12 4AH.

TPC TRAINING SYSTEMS (Volume 7)

A. Dun and Bradstreet Company, 1301 South Grove Avenue, Barrington, Illinois, 60010. U.S.A.

VARIOUS MANUFACTURERS INSTALLATION, START-UP AND SERVICE MANUALS AND BULLETINS.

and many other publications.

SECTION II.

WORKSHEET/ASSESSMENT SHEET

1. Using the Low Velocity Fan Unit in the air conditioning laboratory, find the supply air volume as follows:

   (a) In the supply duct upstream of the fan.

   (b) Traverse across the filter bank.

   Fill in and complete the *Duct Velocity* and *Filter/coil velocity* traverse worksheets on the following pages.

2. Using the Low Velocity Fan Unit, find:

   (a) The static pressure across the fan. *Show all calculations and describe the procedure.*

BARBER COLMAN

unifo® Engineered

AIR BALANCING WORK SHEET

Filter/Coil Velocity Traverse

Filter/Coil Size _____ Date _____ Test No. _____

Job Name _____

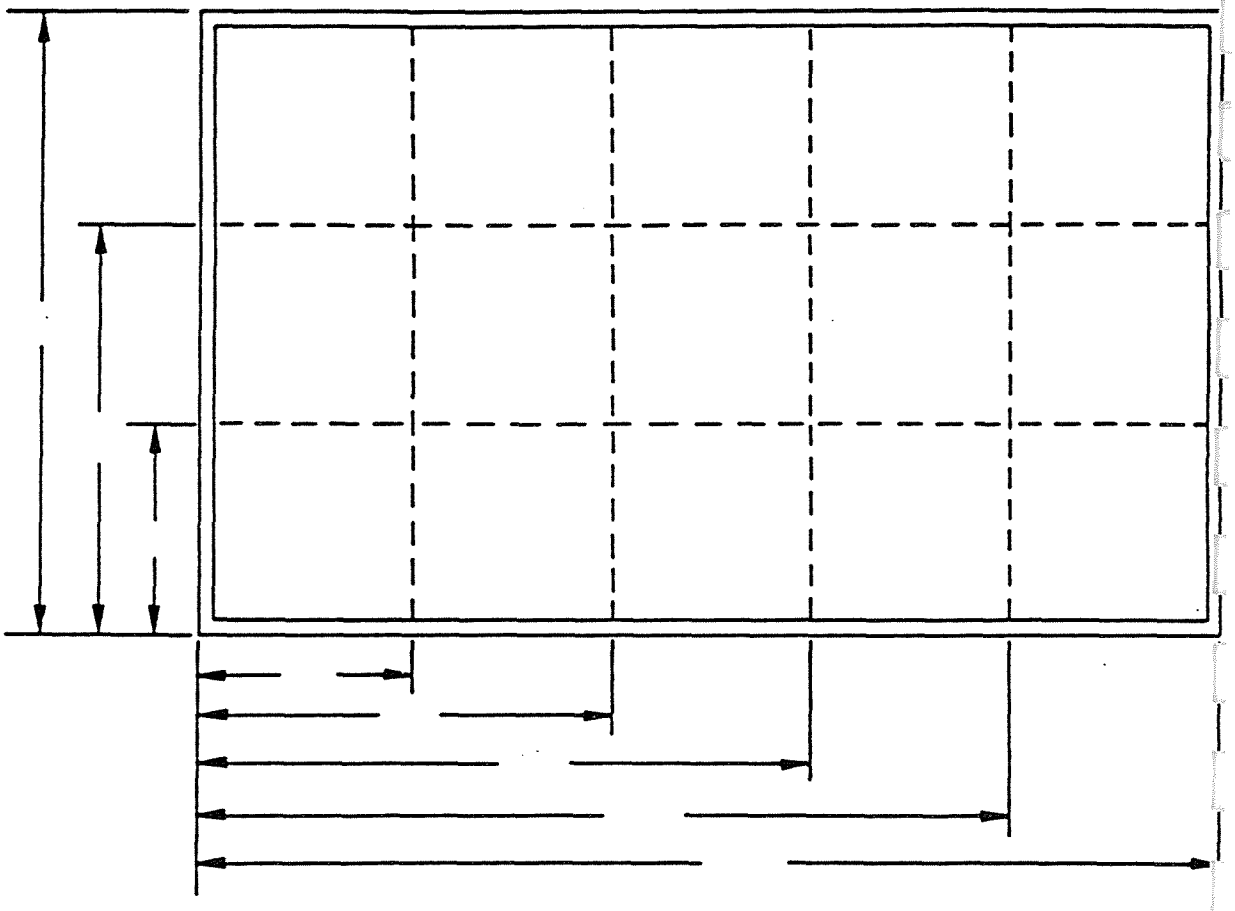
Filter/Coil Area _____

Location _____

Design m^3/s _____

System _____

Actual m^3/s _____



Area (sq. m.) X Average Velocity (m/s) = Air Volume (m^3/s)

_____ sq. m. X _____ m/s = _____ m^3/s

Area Average Velocity Air Volume

Instrument _____

By _____

AIR BALANCING WORK SHEET

Duct Velocity Traverse

Barber-Colman Company
Air Distribution Division
1300 Rock Street
Rockford, IL 61101

Date_____ Test No._____

Job Name_____

Location_____

System_____

Duct Size_____

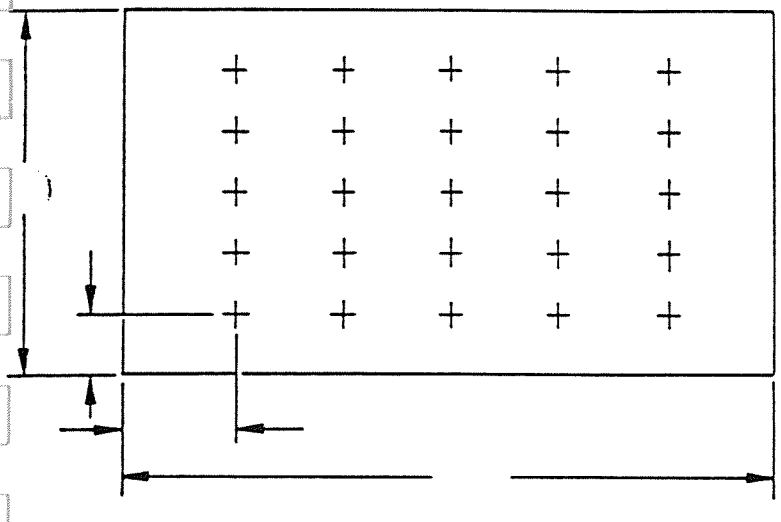
Duct Area_____

Design m^3/s_____

Actual m^3/s_____

Traverse Points	DUCT VELOCITIES - m/s			
	Traverse			
	1	2	3	4
1				
2				
3				
4				
5				
6				
7				
8				
9				
10				
11				
12				
13				
14				
15				
16				
17				
18				
19				
20				
21				
22				
23				
24				
25				
Average Velocity				
Duct Area				
Total m^3/s				

Layout



Instrument

By

(b) Having found the static pressure across the fan, how do you use this to find the fan volume?

(c) How do you check the volume flow rate found in (b)?

3. On a system nominated by your teacher, find the volume flow rate from any three (3) outlets.

(a) What instruments did you use?

(b) How did you calculate the flow from each outlet?
Show all calculations - detail any information you feel must be supplied before calculations can be assumed as correct.

(c) Fill in the worksheet section following for the three outlets.

BALANCING BY_____ INSTRUMENT_____

FAN DATA: RPM_____ CFM_____ S P._____ MOTOR AMPS_____ VOLTAGE_____

1	2	3	4	5	6	7	8
ROOM LOCATION	Supply or Return Model Size	Flow Factors or Net Core Area m^2	Design Air Flow L/S	Design Outlet Velocity m/s	Average Velocity Reading — m/s	Measured Air Flow L/S	REMARKS

(d) Close off *one* outlet and then re-calculate the flow rate of all three.
Evaluate the results.

_____

_____

_____

_____

_____

_____

_____

4. Using the Low Velocity Fan Unit you will be given a system sketch for the air distribution system.
The teacher will reduce the air flow rate by closing the main coil damper.

(a) Adjust all registers and diffusers using a proportional balancing method.

(b) Open the main coil damper to bring supply air up to actual requirements.

(c) From your own observations, does the proportional balancing system appear to work?

5. Balance the same system using the *Balance from the fan out* method. Which method is the quickest?

6. Describe how you would arrive at the required mixture temperature when balancing an air conditioning system.

8. (a) If using a water manometer and it is calibrated in Pascals, how would you convert the manometer reading to a velocity reading in m/s?

(b) Assuming the reading in (a) is 12 Pa, what is the duct velocity in m/s?
Show formula and all calculations using Pa.

(c) What would be reading (b) if the manometer is calibrated in mm?

What is the duct velocity in m/s?
Show formula and all calculations using mm.

NOTE: This section has been SATISFACTORILY completed.

DATE: _____

INSTRUCTOR/S: _____

Student mark for this section is:

60	70	80	90	100

THE MARK FOR THIS UNIT IS:

$$\square + \square = \frac{\square}{2}$$

$$\therefore \quad \boxed{\square}$$

VENTILATION NR13

Learning Outcome 6

AIR HANDLING SYSTEMS

In order to comply with Sections 45 and 46 of the Public Health Act 1991 in respect of the installation and maintenance of air handling systems, the following requirements must be met.

- All drainage and liquid discharges are to be dicharged into a waste water system, or otherwise disposed of, as approved by the relevant public authorities.
- Supply air filters shall be installed on air handling systems (see Appendix 6 for minimum standard).
- On completion of installation, and before being brought into service, the system is to be cleaned.
- Outside air intakes and exhaust outlets must be inspected montly.
- Any maintenance work found to be necessary as a result of the monthly inspection of air intakes and exhaust outlets is to be carried out prior to the next inspection.
- Line strainers, valves, sparge pipes, spray nozzles, and components discharging moisture into the airstream within humidifiers are to be inspected monthly and any necessary maintenance work is to be carried out prior to the next inspection.
- Tanks, trays and discharge devices within humidifiers are to be inspected monthly and any necessary maintenance work is to be carried out prior to the next inspection.
- If an air handling system or a component of an air handling system is shut down on a seasonal basis, it is to be inspected immediately after the shut down and any necessary maintenance work is to be carried out within a reasonable time prior to the next inspection.
- The following parts of an air handling system are to be inspected annually and cleaned if the inspection discloses this to be necessary:
  - coils, trays and sumps
  - condensate drains, tundishes and traps
  - duct work in the vicinity of moisture producing equipment and at access points in the vicinity of fire dampers.
- After condensate drains, tundishes or traps are cleaned, all drainage lines are to be flushed.

Drainage Liquid Discharges

Always a source of problems - not designed correctly - not installed correctly - nil or limited access - Plug Tee pieces should be used. S trap water seals too small (allow double static) - provisions to keep S trap seal from drying out - sized for required when Coil Cleaning.

Condensate Trays

No pools or puddling permitted. Slime build ups must be removed as microbes and fungi would be present. Probably not Legionnaires' as temperatures are too low to support growth - even in winter cycle. Make trays large enough in plan area to carry over - watch design velocities (500 F/M - 2.5 M/S). Negative fall to drain connection. Pipe connection to tray must not stand proud, ie. full drainage is restricted.

NB, Pipework and bottom of trays on cooling coils "Air On" side should be treated to stop condensation.

Coils

Any cooling or heating coil accumulation dust (organic) material, obviously has an inadequate filtration system upstream. Coils which have a spray cycle (for dew point control or other reasons) act as an efficient air filter, therefore require more maintenance and servicing than a "Normal" coil. Whilst working downstream of the cooling coil, when the system is operating don't forget your own safety and wear a mask.

Outside Air intakes - Exhaust Outlets - Return Air Grilles

Legionella may travel over 1 km under certain conditions. There is a lot of commonsense involved here, as understandably the rules have a wide base, "Dirty" air exhaust systems (toilets, car parks, kitchen, cooling towers, etc.) must be located downwind of the prevailing weather pattern.

Consider separate high efficiency filtration to fresh air systems (booster fan?). Keep filter media moisture free to minimise bacteria growth.

Return air grilles and ductwork are an ideal place for bacteria sporing to occur - especially in high population/smoking areas. A constant temperature and organic material is supplied all year for such growth as in restaurants. Similar but to a lesser degree occurs with paper dust, lint etc. Cigarette nicotine adheres to the duct lining, building up with organic food, etc.

Access panels in ductwork and fans for cleaning are most important, but just as important are good lighting and safe access. Again, wear a mask.

Filters

Legionella is only 3 micrometres (Microns) long by 1 Micron in diameter, however, it joins to Protozoa (5-100 Microns) dust.

(1-100 Microns) or Microbes. The filter then needs to remove a much larger particle than the virus itself.

Test Dust No.2 - Particles in 3-10 Micron range are collected (dirt and droplet particles), however, it is no good for viruses (say 0.03 to 0.8 Microns).

Some examples of viral infections are Smallpox, Chickenpox, Measles, Mumps, Influenza and the common cold. The common cold are produced during sneezing, the droplet nuclear are 4 Micron approx.

The better the filter efficiency, the better for the occupiers.

For correct maintenance and installation edge and face sealing is critical. Always seal on the "air on" side, even if an access panel is required. Ensure that all moisture is eradicated from filters and ductwork as bacteria, moulds, fungus and other micro organisms rely on moisture for their survival and growth. They are aquatic life forms.

Humidifiers

More prevalent overseas, not so much in our climate. However, when used in industry, eg. printing works, textile factories and woodworking shops there is an abundance of food for Microbes (cellulose material).

Steam is preferred and "clean steam" (free of chemicals is best). Conditioner and duotwork condensate trapping - access for ease of cleaning - chemical disinfection to be carried out after manual clean if slimes, etc. present.

Water type, whilst more economical and simpler to install for smaller systems, etc. computer rooms, have problems of solids, chemicals and other matter remaining. The end of main seasonal usage (ie. Winter) these systems require vigorous and intensive service and clean up.

WATER COOLING SYSTEMS

The excess heat energy extracted from a building by an air conditioning system is released into the atmosphere via a cooling tower. Because of the temperature of the water and the presence of sludge, algae and so on, the water cooling system may aid the proliferation of *Legionella* and other bacteria.

The aerosol drift from a contaminated tower may cause an infection in nearby susceptible people. The correct operation and regular routine maintenance including cleaning are important to ensure the desired performance and acceptable hygienic conditions of all components of the air conditioning system.

Water Treatment

A water treatment program is essential for a condenser water cooling system to inhibit corrosion, the build-up of scale and the development of any microbial contamination. Corrosion and scale development can cause fouling of the condenser tubes and the

pipework distribution system resulting in poor system efficiency and premature failure. These factors can also provide an environment which promotes the colonisation and growth of organisms such as *Legionella.*

Specialist water treatment companies will be able to advise on the appropriate regimen, whether it is a chemical or non-chemical process or a combination of water treatment processes. These companies should also be familiar with any legislation applicable to the particular products or processes they offer.

Biocides

Water treatment biocides are used to control the growth of bacteria, algae, protozoa and fungi in the condenser water. These micro-organisms may provide nutrients for growth of *Legionella.*

The effectiveness of biocides may be reduced by the presence of organic and inorganic materials such as sand, dirt and other particulate matter.

Biocides must never be allowed to discharge into surface draining systems or other water courses. Approval for discharge in sewage reticulation systems must be obtained from the relevant authority.

Filter and Separators

The use of a correctly designed system of either mainstream or sidestream filtration and/or centrifugal separation can significantly reduce fouling of cooling water systems with particulate matter. This cleansing process allows the biocide or other treatment process to be more effective.

Routine Cleaning

The regular cleaning of cooling towers and any associated condenser water system is important in a well-maintained system.

Such cleaning also reduce the nutrients and microbial populations which may aid in the growth of *Legionella.*

Stagnation of water must be avoided, as this can be conducive to the growth of *Legionella.*

Legal Requirements

In order to comply with sections 45 and 46 of the Public Health Act 1991, in respect of the installation and maintenance of cooling towers, the following requirements must be met.

- All drainage and liquid discharges are to be discharged into a waste water system, or otherwise disposed of, as approved by the relevant public authorities.

- All cooling tower systems are to be inspected monthly.

- All maintenance work found to be necessary as a result of the inspection is to be carried out within a reasonable time prior to the next inspection.

- All systems are to be cleaned at three montly intervals. Approval to extend the three-monthly cleaning interval may, in individual cases, be given by the Director General of the NSW Health Department.

- If a cooling tower system is operated on a seasonal basis, it is to be drained as soon as practicable after shut down. Where it is incapable of being shut down, water treatment shall be maintained.

- Cooling towers operating on a seasonal basis are to be inspected and any necessary maintenance work carried out prior to recommissioning.

HOT AND WARM WATER SYSTEMS

Hot water systems operate at temperature of 60°C and above, measured at the outlet.

Warm water bathing system, which are designed to prevent scalding, operate at a temperature of 35 to 43.5°C measured at the outlet. These temperatures are conducive of the growth of *Legionella* and other micro-organisms. The warm water may be generated instantaneously with the aid of approved fail-safe type thermostatically controlled hot and cold water mixing valves. These systems are widely used in health care facilities such as hospitals and nursing homes.

Stratification of water temperature can occur in water heaters and warm and hot water storage tanks. The temperature within the vessels can result in *Legionella* proliferating.

Water Treatment

Because hot, warm and cold water are classified as potable water (fit for drinking), there are limits to the water treatment that can be applied to control any growth of *Legionella*.

Routine disinfection by chlorination to maintain a low level of free residual chlorine is one of the few proven methods. The method must be accurate, automatic and filtration may be a prerequisite.

So far, other methods for application in large warm water systems have shown to be unreliable and unacceptable.

Other Water Treatments

Microbial growth in warm water systems can be controlled by various non-chemical treatments. However, the efficacy of these treatments has not been fully determined in the field under varying conditions such as water quality, temperature of storage, volume of water use and system design.

Two methods of microbial control in warm water system which have potential application are using the effects of heat and ultra-violet irradiation.

Ultra-violet light treatment of warm water systems which have potential application are using the effect of heat and ultra-violet irradiation.

Ultra-violet light treatment of warm water appears at this stage to have limited application.

EVAPORATIVE AIR COOLERS

Evaporative air coolers operate by utilising the physical phenomenon of cooling "air" entering the building by evaporation of water.

These systems will only work satisfactorily in dry climates as a high level of humidity prohibits an adequate rate of evaporation. They are used in large numbers in areas remote from the coast, for instance in western NSW. At this stage they have not been implicated in any outbreak of Legionnaires' disease, although *Legionella* bacteria have been found in such systems.

Evaporative air coolers required regular attention and manufacturer's instructions for operation must be followed.

Minimising Contamination

Before switching the unit off, the fan should be allowed to dry the filter pads.

Evaporative coolers should be fitted with a bleed-off system. This is essential to prevent excessive accumulations of dissolved solids and other impurities within the unit.

Maintenance of Systems

All systems shall comply with the following general maintenance requirements.

Procedures to be Taken During Maintenance

If maintenance of a regulated system is being carried out on the premises on which it is installed, the contractor or employee carrying out the maintenance is guilty of an offence if appropriate measures are not take to:

- help prevent or minimise adjoining areas and the ambient environment from being contaminated by aerosols or the generation of dust or particulate matter; and
- prevent public access to the area in which the maintenance is being carried out.

Maintenance Records

- Whether a maintenance inspection of a regulated system is carried out, the occupier or maintenance contractor responsible for that plant shall make a written record of the date and details of the inspection.
- Whenever maintenance work is carried out on a regulated system, the occupier for

maintenance contractor responsible for the plant shall make a written record of the date and nature of the work performed and the name of the employer. The maintenance record is to be signed by the person who actually performs the work.

- An occupier or maintenance contractor who fails to make a written record is guilty of an offence.

- Any person who removes maintenance records from premises within 12 months after the record is made is guilty of an offence.

- An authorised officer may enter any regulated premises and inspect the records.

SAFE WORKING PRACTICE

Chemicals used in operating, maintenance and cleaning procedures must be treated with caution. Legionellosis is, of course, also a potential health hazard for people who work around systems harbouring the bacteria. Safe working practices are vital and it is important that safety measures are instituted as soon as possible.

Safety measures must be observed by all, whether they are a maintenance worker, a building inspector or someone taking a sample from the system. The risk is from contaminated aerosols and spray mists, as well as the obvious hazards of working with chemicals and around structures with difficult or inadequate access.

New installations must be designed and constructed to provide safe access, while existing installations must be made safe without delay.

Protective measures must be taken during maintenance and inspections of air handling and water cooling systems to reduce the risk of inhalation of spray mists or exposure to toxic chemicals.

Respirators must comply with AS1716 and be used in accordance with AS1715. The minimum respiratory protection shall be provided by a Class M half facepiece particulate filter.

The location of the system must be taken into consideration to ensure that maintenance and cleaning activities do not put any persons or adjoining premises at risk. AS2645 gives guidance to precautions to be observed when working in confined spaces such as cooling towers or storage tanks.

Anyone taking a sample from a system, particularly during a suspected outbreak, must also take protective measures. If possible, the system should be turned off before a sample is taken.

A Class L particulate filter is not acceptable as adequate protection against contaminated aerosols. In all other respects Table A1 of Appendix A of AS3666 can be considered as the minimum requirement for personal protective equipment during maintenance of air handling and water systems.

BUILDING CONSTRUCTION AND MODIFICATIONS

Fresh-air intakes must be located away from cooling towers and exhaust discharges from air handling system to avoid cross contamination from the same or nearby buildings. Prevailing wind directions should also be considered. Air intakes must be designed and installed to minimise the entry of rainwater and prevent the entry of birds, rodents and windblown material such as leaves and paper. They must be of a sufficient height above the ground or surrounding rod to minimise the intake of dust and detritus.

Cooling towers must not be located near exhaust discharges from kitchens or other areas where nutrients conveyed in these systems could assist in the growth of *Legionella*.

When considering the relocation or repositioning of duckwork in a building, care should be taken to ensure that the duct work is designed and installed to minimise the ingress and accumulation of moisture. This includes grading ductwork to prevent water collection. As an added precaution, all ductwork must be cleaned before the air handling system is commissioned.

Locations near occupied areas, pedestrian thoroughfares, air intakes and building openings should be avoided.